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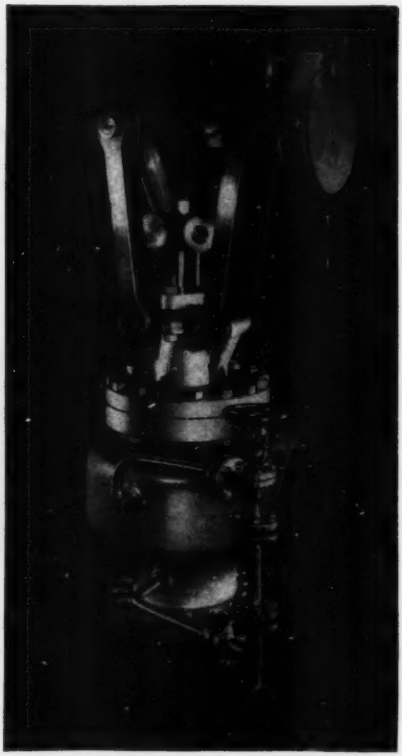

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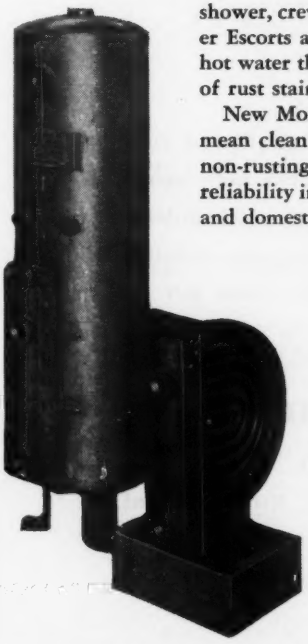

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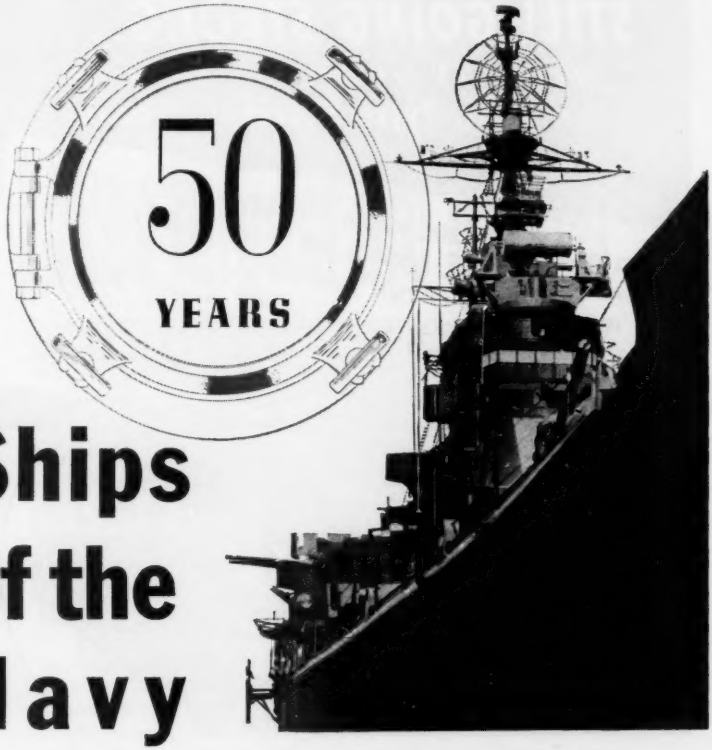

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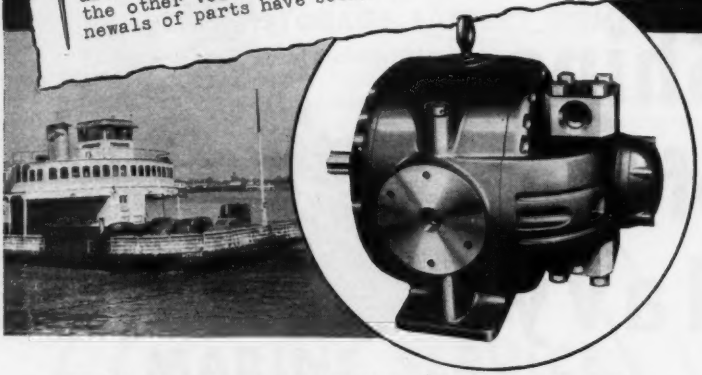

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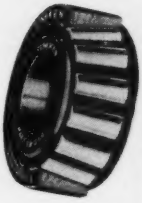

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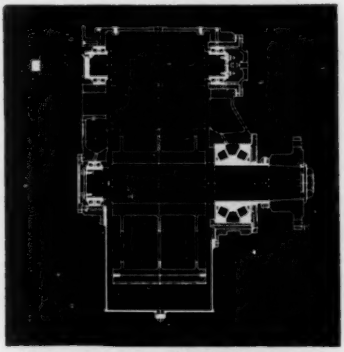
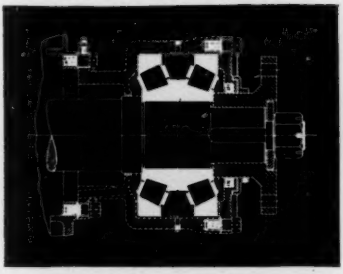
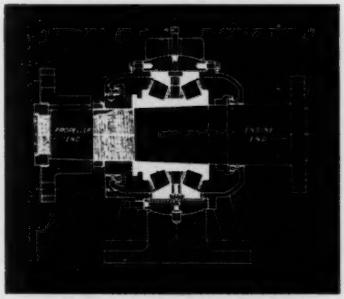


Single reduction herringbone gear marine drive, completely equipped with Timken bearings. Thrust taken on large Timken bearings at stern end of main shaft.



Timken marine thrust bearing mounted on an extension of the engine crank shaft and bolted to the engine case. Lubricated by engine oil.



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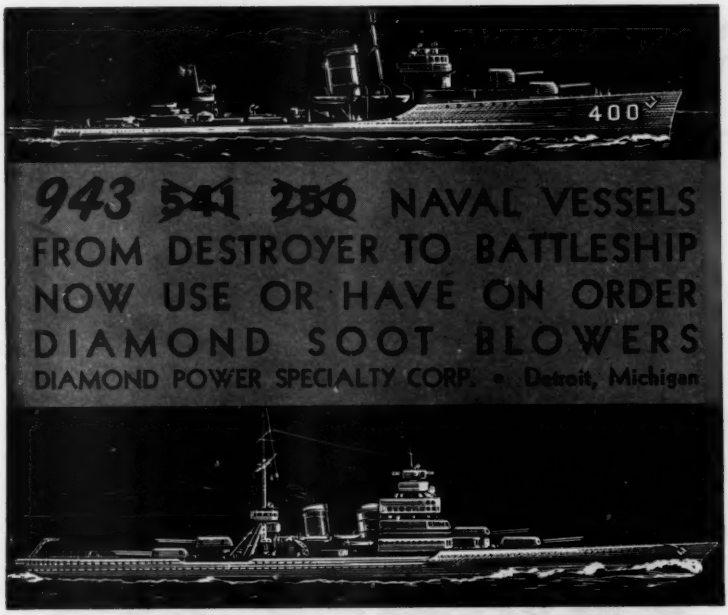
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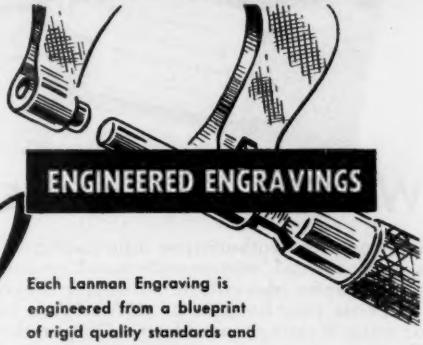

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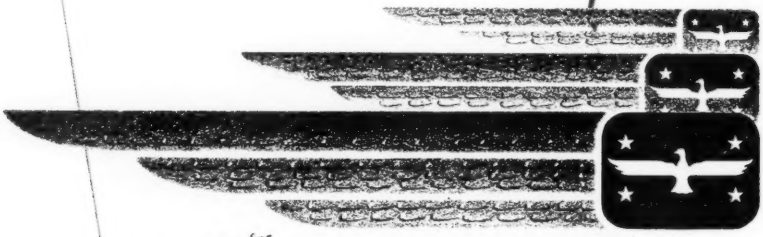

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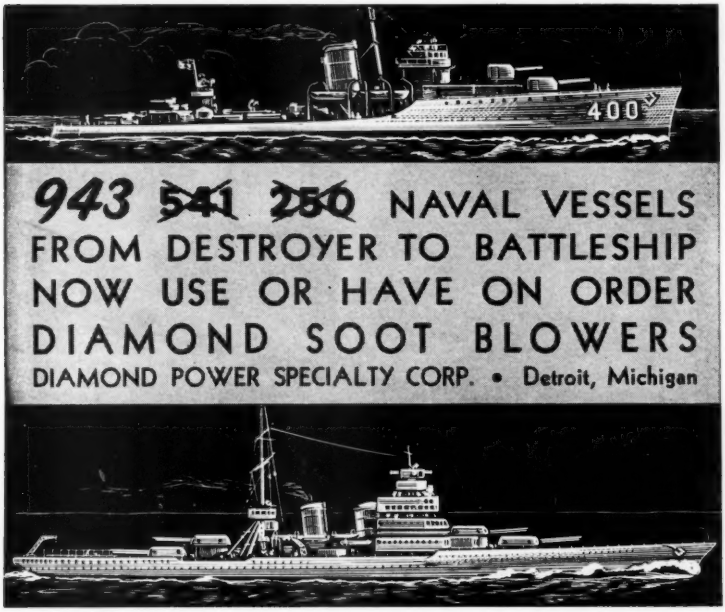

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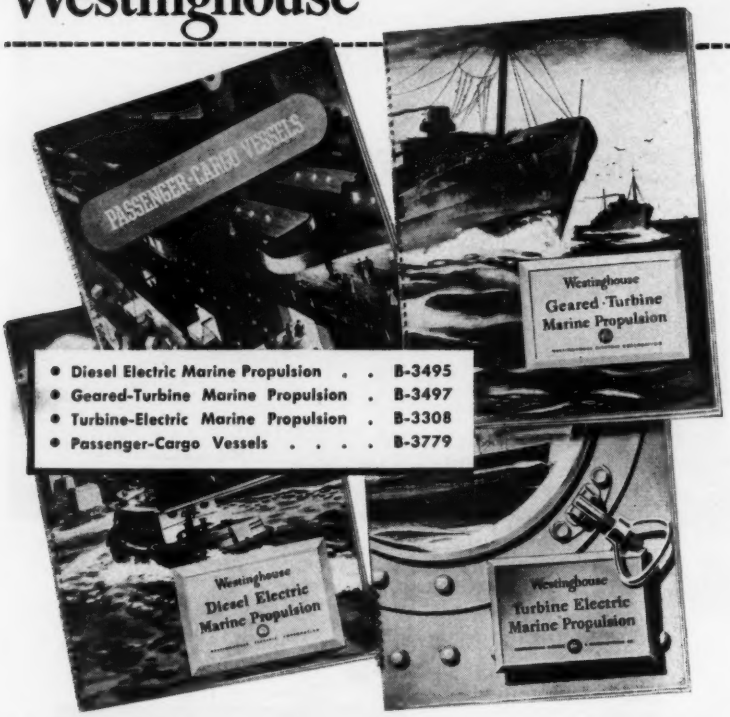

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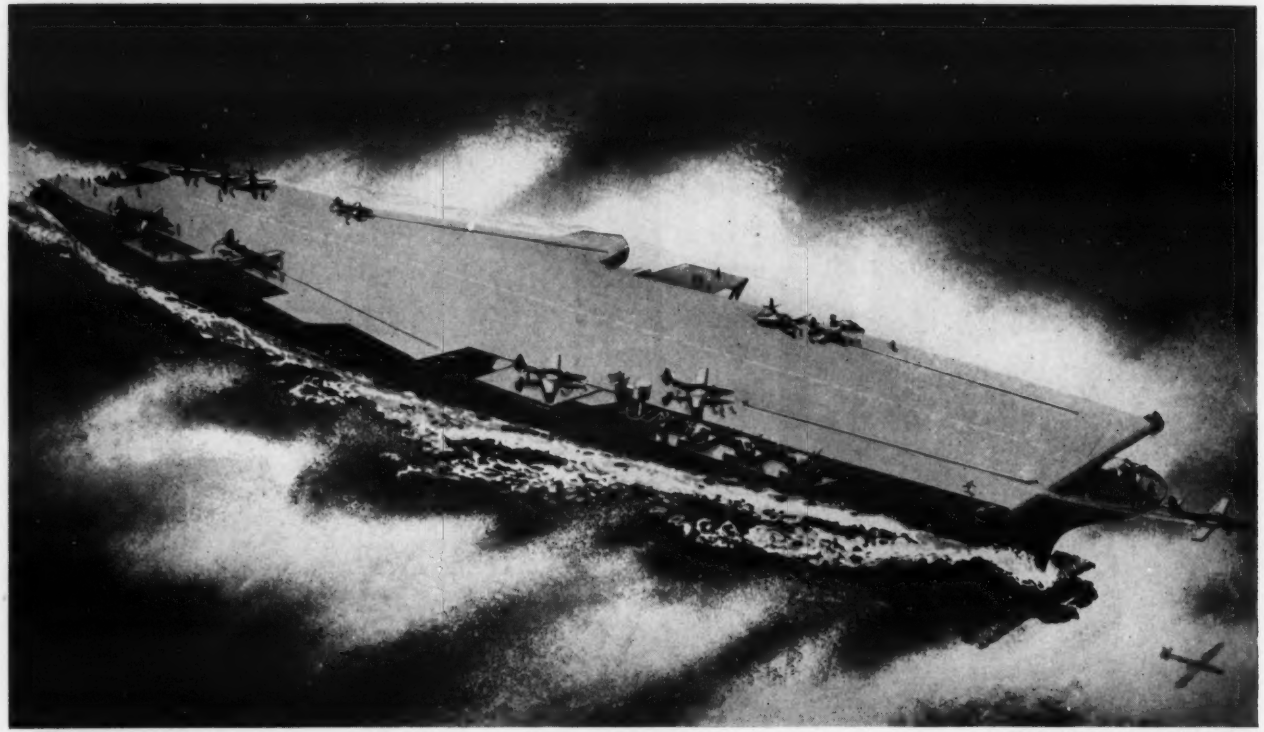

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OF

THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.

VOL. 61. AUGUST 1949. No. 3.

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SECRETARY'S NOTES.

The Society operated during 1948 at a net loss somewhat in excess of $10,000. This was expected as previously has been reported. It is hoped that gradual developments will improve this situation so that there will be no need to resort to extreme action a few years hence.

Members' action to bring in new members has been very satisfactory as evidenced by the larger than usual lists published in the last two issues. If we are to approach the break-even point it will be necessary to continue this drive indefinitely. With the large number of eligibles, it should be possible for the Society to continue to grow at the present accelerated rate for some time to come.

An indirect result of expanded membership could be an increase in the volume of advertising. Campaigns to obtain new advertisers are contrary to past policy of the Society and it is desired to continue this policy. However, increased circulation of the

JOURNAL will increase its present high value as an advertising medium and may attract new advertisers without a drive. Any increase of income from this source will advance our progress toward a sound fiscal condition.

A restatement of the Society's present operating policy is submitted for the information of members. It is:

To hold the present rate of annual dues at $5.00.

To maintain and improve the technical value of the JOURNAL, both qualitatively and quantitatively.

To improve the Society's income by obtaining a gradual accession of new members with the hope that this may also result in increased advertising revenue and

To operate with a minimum annual loss until either the break-even point is passed or the policy has to be re-cast because of the treasury balance.

There has been no response to an earlier request that photographs of interest be submitted for publication.

There has been no response to an earlier request for obituaries of deceased members.

There has been a very limited response to an earlier request for original manuscripts or for suggested titles for articles.

Surely the members of the Society have not joined that group which feels that all knowledge and wisdom are centered in Washington.

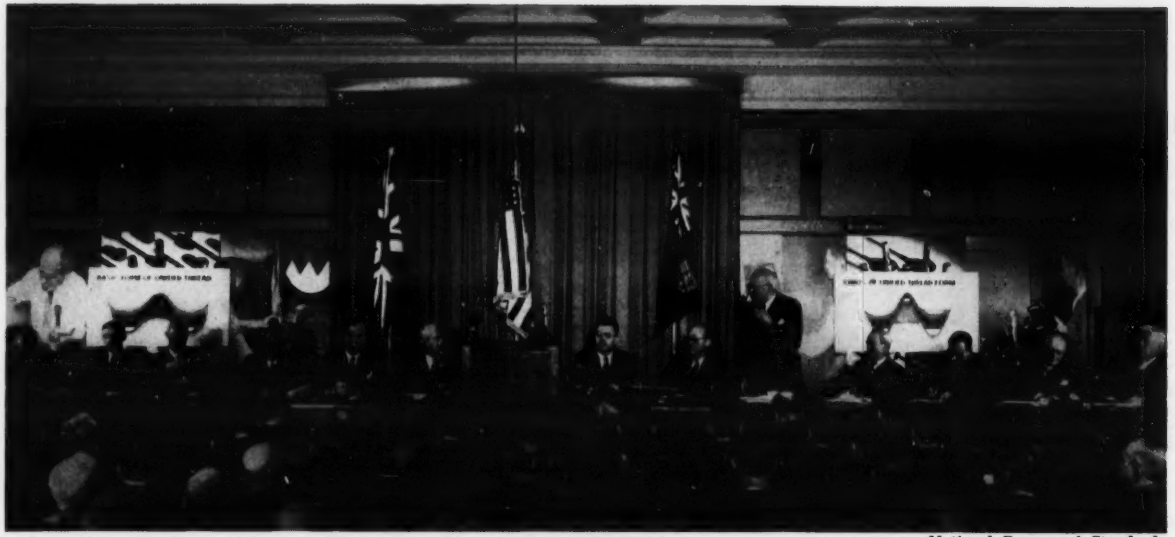
PART ONE

ORIGINAL ARTICLES

The articles which the Editor, acting under the supervision of the Council, has accepted for publication in Part One are the original works of the authors.

They are offered in furtherance of the Society's raison d'etre: to further the advancement of naval engineering. Discussion of any article is invited for publication in a subsequent issue.

Permission to reprint is freely granted on condition that *full acknowledgment is given* to the author and to the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.



National Bureau of Standards.

Delegations from the three countries at the speakers' table at ceremony marking agreement on unified system of screw threads.

Left to right: Percy Good; T. R. B. Sanders; Sir Ewart Smith; C. D. Howe; Sir Oliver Franks (Ambassador of Great Britain); Charles Sawyer (U. S. Secretary of Commerce); Dr. Edward U. Condon (Director, National Bureau of Standards); Hume Wrong (Ambassador of Canada); Howard Coonley, (chairman, Executive Committee, American Standards Association); Donald Carpenter (chairman, Munitions Board); William L. Batt; James G. Morrow; David R. Miller (Interdepartmental Screw Thread Committee). Frank P. Tisch, Vice-Chairman, BI Sectional Committee, does not appear in the picture.

INTERNATIONAL SCREW THREAD STANDARDIZATION — Presentation of the procedure for taking a pioneer step in a field of great importance to engineering.

BY J. W. JENKINS, MEMBER AND K. D. WILLIAMS, MEMBER.

THE AUTHORS.

J. W. JENKINS attended Case Institute of Technology for 3 years, transferred to and graduated from University of Wisconsin in 1917 with the degree B.S. He served during World War I as a field artillery, and (later) as a flying officer (A.S.M.E.). He resigned in 1919 to enter on 15 years of industrial engineering experience including consulting work involving specification, design, and application of engineering materials. He joined the force of the Inspector of Naval Materials, Chicago in May 1934 serving as Supervisor of the South Bend District and as technical assistant to the Inspector of Machinery, Beloit, Wisconsin. He joined the Bureau of Ships in July 1940 and has since served in the Metallurgical and Mechanical Branch of the Research Division, presently as the Principal Civilian Engineer of this Branch. He has for the past 8 years carried out standardization matters for the Branch and has served as liaison with the Standards Branch.

K. D. WILLIAMS graduated from Kenyon College with the degree of Ph.B. in 1903. He was employed in the forging department of Midvale Steel Co., Nicetown, Pa. from 1905 to 1910. In 1910 he accepted appointment as an assistant inspector of engineering material in the office of Inspector of Engineering Material U.S.N., Chester, Pa. In April of 1917 he was transferred to the Bureau of Steam Engineering, Specification Division where he was the Chief Metallurgical Engineer. During the subsequent 32 years he has continued, under the Bureaus of Engineering and Ships to effectively guide the development and standardization of metallurgical and mechanical materials. His war work won him acclaim in the form of the Navy's Distinguished Civilian Award, The American Society for Metals citation for distinguished service to industry and the American Standards Association citation for his work in establishing war standards. Mr. Williams retired on 31 March 1949.

Agreement on Unification.

On November the 18th 1948, at the National Bureau of Standards of the United States, in the city of Washington, D. C. was signed a Declaration of Accord with respect to the Unification of Screw Threads. Distinguished delegates from Canada and the United Kingdom as well as high officials of the United States took part in the celebration and signing of the Agreement. The work provided that standards for the unified screw threads in the publications of the British Standards Institution, the Canadian Standards Association, the American Standards Association, and the Interdepartmental Screw

Chairman, BI Sectional Committee, does not appear in the picture.

Declaration of Accord

with respect to the

Unification of Screw Threads

It is hereby declared that the undersigned, representatives of their Government and Industry Bodies, charged with the development of standards for screw threads, Agree that the standards for the Unified Screw Threads given in the publications of the Committees of the British Standards Institution, Canadian Standards Association, American Standards Association and of the Interdepartmental Screw Thread Committee fulfill all of the basic requirements for general interchangeability of threaded products made in accordance with any of these standards.

The Bodies noted above will maintain continuous cooperation in the further development and extension of these standards.

Signed in Washington, D. C., this 18th day of November, 1948, at the National Bureau of Standards of the United States.

le D. Howe	Ministry of Trade and Commerce, Dominion of Canada
J. S. Morrow	Canadian Standards Association
T. R. B. Sanders.	Ministry of Supply, United Kingdom
Percy Good Ernest Smith	British Standards Institution
	Representative of British Industry
E. U. Condon	National Bureau of Standards U. S. Department of Commerce Interdepartmental Screw Thread Committee
Fred P. Peters	American Standards Association The American Society of Mechanical Engineers Society of Automotive Engineers
William L. Batt	Sponsors Council of United States and United Kingdom on the Unification of Screw Threads

Thread Committee will fulfill all of the basic requirements for general interchangeability of common threaded products.

This was no minor or easily accomplished objective nor can the results of this unification be evaluated on any but the highest plane of importance. A true evaluation must recognize not only the immediate accomplishment however important. It should as well recognize that the Accord marks the beginning of and provides impetus to the realization of unification on a much broader basis. It points the way ultimately to fulfilling the need for interchangeability of parts of manufacured products thus leading to fully developed trade and commerce among the signatory nations.

Brief Historical Considerations.

The man on the street may not have viewed with concern the fact that since 1845 the British have had one thread form while since 1865 we have had a different one. This lack of interchangeability may not have been significantly important during our early industrial history.

However, World War I brought about joint Military and Industrial effort by the United States and Britain and, with it, a serious problem in lack of such interchangeability. Efforts toward unification resulted. The Congress, in recognition of the importance of the problem, in July 1918, authorized the appointment of the National Screw Thread Commission to investigate and promulgate standards for screw threads. In 1919 after a thorough study of screw thread practice in this country this commission conferred with British and French Engineers and Manufacturers with the hope that these findings might form the basis for international standardization. Discussions ranging up to 1926 were finally dropped as not likely to be successful. Natural conflicts of interest were apparently insurmountable in the face of the world wide sense of security engendered by industrial status quo.

World War II demanded of American Industry a great volume of equipment with British threads. American Forces based in England used vast quantities of American threads at times needing immediate replacement. Neither industrial set-up was prepared or tooled to meet the problems and this fact led to very costly and extremely serious delay.

The issue of a Unified Standard was indeed revived in extremely aggravated form.

Canada was also involved. The Hon. C. D. Howe, Minister of Reconstruction and Supply and Canadian Member of the Combined Production and Resources Board emphasized this aggravated problem when he said: "Perhaps we in Canada realize the need for standardization more than any other country because we are sort of in between British Standards and Standards of the U. S. Many of our designs are British designs and a great deal of our equipment is U. S. equipment built to the Standards of the U. S."

In 1943 the first of a series of meetings under the auspices of the Combined Production and Resources Board and involving the U. S., Canada, and the United Kingdom, was held in New York City. A second meeting was covered by the "Report on Conferences on Standardization of Screw Threads and Cylindrical Fits, London, August-September 1944".(1) A third meeting was covered by "Report of Conference on Unification of Engineering Standards—Screw Threads, Pipe Threads, Limits and Fits, Drawing Practice and Metrology—Ottowa, Canada, September-October 1945".(2)

As indicated by the titles of the Conference Reports, progress toward unification was in the making.

At Ottowa, the British made a concession which perhaps was the key to the success of the whole effort. This action consisted of abandonment of the traditional British 55° thread angle in favor of the American Standard of 60°. The three principal characteristics of screw threads are (1) Angle and form of thread, (2) Number of threads per inch for the various series of thread diameters, and (3) The tolerances and allowances for the classes of thread fits. Accurate fits between components having different thread angles were impossible.

By this concession which had to be made with due regard to the impact on the British economy, the way was cleared for unification. It was now practicable to attack the vast amount of detailed work which remained to be done in developing and agreeing upon classes or grades of fit and corresponding tolerances and allowances.

There were two additional main and several semi-formal international conferences which reflected the work that was going on behind the scenes and leading up to the development of the Unified Standards and to the final ceremony of the signing of the Declaration of Accord.

Mechanism.

This paper will fail in its purpose if we cannot with some clearness reveal the behind the scenes mechanism by which the many and conflicting interests were persuaded, adjusted, or compromised so that final actions could be smoothed into acceptable form and result in simplification and standardization accord.

Over the five years leading up to November 1948 both preceding and following the main meetings mentioned above, there were countless committee meetings to develop details.

The detailed work was carried on by Committees, Sub-committees, Task Groups, and by individuals whose interest in many cases led them to effort beyond the normal scope of their duties that progress might prosper. These people were seasoned engineers, technicians, even executives. They represented manufacturers, consumers, standards associations, and the various Government Agencies. They were selected at the technical level of the operating unit whether Industrial or Government where compromise and adjustment is possible. They were selected because of their broad knowledge not only of the immediate subject but of the ramifications and effects of their judgments.

All of the natural conflicts of interest were present. Perhaps no more perfect setup ever existed for the rugged individualist to exercise his inalienable rights and thereby to create a traffic jam and traffic jams there were.

It is safe to say however that there were few among the many conferees who did not recognize the necessity for cooperative effort. It was recognized that the success of any useful undertaking is dependent upon full understanding of its objectives and advantages on the part of those whose cooperation and support are essential to its proper development.

Many minor conflicts of interest had to give way to the accomplishment of accord. Clashes occured over the objectives and effects of gaging practices. Much persuasion was needed to effect separation of special threads (a tremendous and additional problem in their own right). Thus the way was cleared for standardization of the thread forms, dimensions, allowances, and tolerances of standard fasteners. Interchangeability between the old and new product and even terminology were battles of no mean size.

In short, the success of these working groups was concommitant with the fact that they were composed of or sup-

ported by technical people capable of recognition of the importance of cooperative effort.

As surely as this agreement was guided to fruition on 18 November 1948 the proceedings of its hard hitting working personnel who combatted inertia and resistance, who compromised natural conflicts of interest and who persuaded cooperative action, exemplified the democratic process in its finest tradition.

It is with regret that individual working membership lists of the British and Canadian Committees cannot be published here. However attendance lists together with affiliations are published in the references at the end of this paper. It is also regretted that no mention of the many alternates to American Committees can be made.

However since it is desirable to briefly review the technical basis of the accord and its effect upon the standards of the Federal Government of the United States as well as to indicate how correlation with Industry standards is effected, the following personnel lists are included.

The Interdepartmental Screw Thread Committee

Honorable Edward U. Condon, Chairman
Mr. K. D. Williams, Vice-Chairman
Mr. I. H. Fullmer, Secretary

Mr. Stanley Farrow	Mr. H. W. Robb	Mr. D. R. Miller
Mr. B. H. Slocum	Mr. W. C. Stewart	Mr. H. C. Erdman
Mr. A. F. Wentzel	Mr. Eugene von Loesch	Mr. P. G. Schulz
Mr. P. M. Delzell	Mr. Frank Richardson	

The Sectional Committee on The Standardization and Unification of Screw Threads.

Mr. Elmer J. Bryant, Chairman
Mr. Frank P. Tisch, Vice-Chairman
Mr. Walter R. Penman, Secretary
Mr. Paul J. DesJardins, Chairman, Subcommittee No. 1

Mr. W. L. Barth	Mr. W. H. Gourlie	Mr. L. F. Polk
Mr. Carl W. Bettcher	Mr. Harry C. Graham	Mr. C. M. Pond
Mr. John S. Black	Mr. H. B. Hambleton	Mr. T. W. Ragan
Mr. C. B. Bronson	Mr. W. M. Hanneman	Mr. F. E. Richardson
Mr. W. G. Brown	Mr. Frederick W. Helming	Mr. H. W. Robb
Mr. W. S. Brown	Mr. Arthur M. Houser	Mr. Paul G. Schulz
Mr. Earle Buckingham	Mr. P. L. Houser	Mr. B. H. Slocum
Mr. G. Carvelli	Mr. H. L. Keller	Mr. S. B. Terry
Mr. George S. Case	Mr. F. C. Leiner	Mr. E. A. Thatcher
Mr. C. G. Davey	Mr. Harry A. Marchant	Mr. G. T. Trundle, Jr.
Mr. P. M. Delzell	Mr. C. F. McElwain	Mr. W. G. Waltermire
Mr. H. C. Erdman	Mr. Henry C. E. Meyer	Mr. A. F. Wentzel
Mr. E. C. Erickson	Mr. D. R. Miller	Mr. C. A. Werme
Mr. Stanley Farrow	Mr. Paul V. Miller	Mr. C. C. Winter
Mr. R. A. Frye	Mr. H. J. Muth	Mr. K. D. Williams
Mr. I. H. Fullmer	Mr. L. F. Parlette	
Mr. H. Fussner	Mr. R. H. Perry	

The above Committees reveal representation of Industrial Consumer, Manufacturing and Fasteners Institute interests on the Interdepartmental Screw Thread Committee of the Government. They also reveal Government Agencies' interests on the Sectional Committee on the Standardization and Unification of Screw Threads of the American Standards Association.

Technical Basis of the Accord.

The three principal characteristics of screw threads are angle and form of the thread, the number of threads per inch for the various series of thread diameters, and the tolerances and allowances for the three classes of thread fits ("loose," "medium," and "close"). The British system, originated by Whitworth in 1845, is based on a thread angle of 55 degrees with a thread form having rounded crests and roots. The American system, developed by Sellers in 1864, has a thread angle of 60 degrees with a thread form having flat crests and roots. The number of threads per inch for the various series of thread diameters was the same in both systems, with the exceptions of the half-inch-size coarse thread. Accurate fits between components having different thread angles were impossible. Moreover, the tolerances and allowances varied in the two systems. The present unification agreement provides a 60-degree angle and a rounded root for screw threads. The crest of the thread may be flat, as preferred in American practice, or rounded, as preferred by the British. The number of threads per inch for the various series of thread diameters has been made uniform, and the limiting dimensions for three classes of fit have been agreed upon. Thus, interchangeability of screw-thread parts, based on the accord, now becomes feasible. The consummation of this accord between the English-speaking nations will be of inestimable value to the armed services of the nations concerned during emergency periods.

The Unified Form of Thread.

a. Angle of Thread: The basic angle of thread between the flanks of the thread, measured in an axial plane, is 60°. The line bisecting this 60° angle is perpendicular to the axis of the screw thread.
b. Form of Crest: The crest may be either flat or rounded.
c. Form of Root: The root of the screw shall have a rounded contour.
d. Illustration: Plate 1 shows the design forms (maximum metal condition) of the external and internal threads of the unified form of thread.

Thread Series.

The thread series will remain with no change except for the symbol. The letter U has been added to denote the unified thread form as follows:

	Old	*New*
a. Coarse thread series.	NC	UNC
b. Fine thread series.	NF	UNF
c. Extra-fine thread series.	NEF	UNEF
d. 8-thread series	8N	8UN
e. 12-thread series.	12N	12UN
f. 16-thread series.	16N	16UN
g. Special threads.	NS	UNS

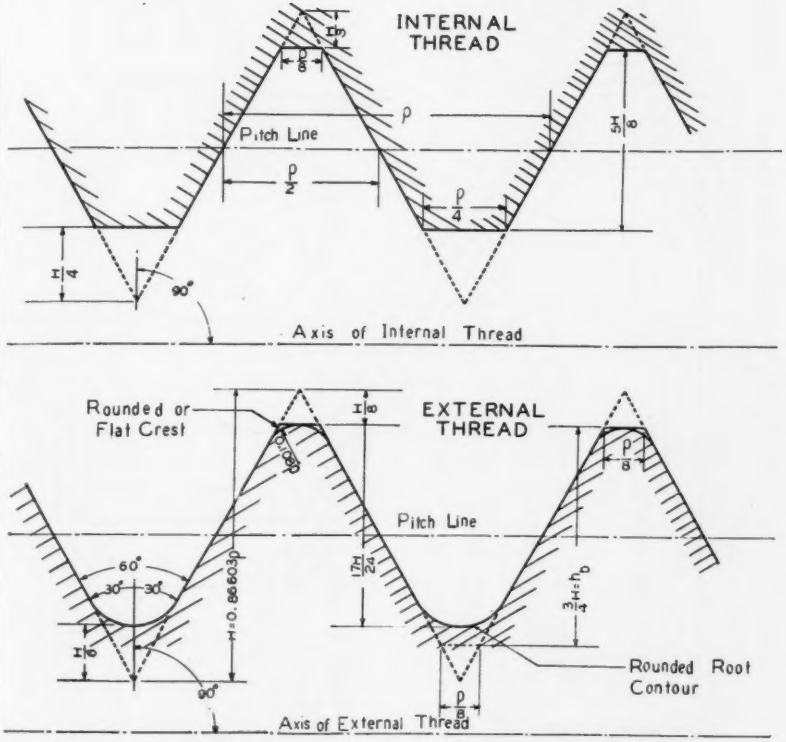


PLATE 1.—Unified Form of Thread (Maximum Metal Condition).

Classification and Tolerances.

There are three distinct classes of unified screw-thread fits established for general use. They are distinguished from each other by the amounts of tolerance and allowance specified. This includes classes 1A, 2A, and 3A, applied to external threads only, and classes IB, 2B, and 3B, applied to internal threads only. Classes 1A and 2A include a positive-pitch diameter allowance; class 3A does not. Plate 2 illustrates tolerances and allowances for classes 1A, 1B, 2A, and 2B. Plate 3 illustrates tolerances for 3A and 3B. Classes 1A and 2A allowances are the same for all sizes and pitches. The maximum looseness of fit that can be obtained under the new standard is greater than at present for all sizes as shown.

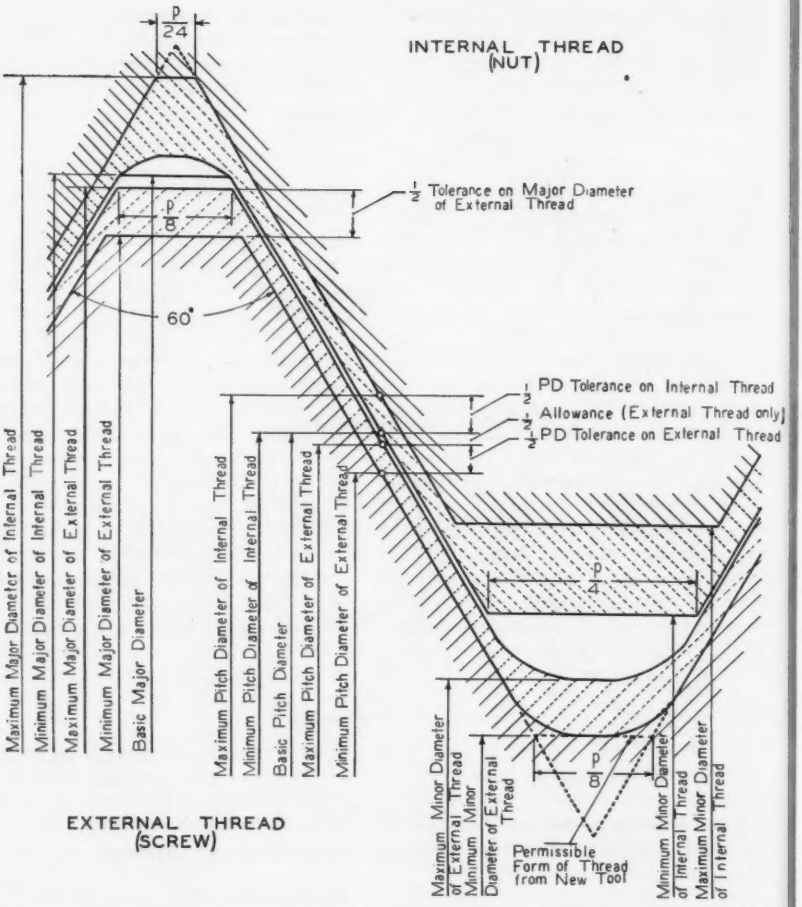

INTERNAL THREAD
(NUT)
P/24
1/2 Tolerance on Major Diameter of External Thread
P/8
60°
1/2 PD Tolerance on Internal Thread
Allowance (External Thread only)
1/2 PD Tolerance on External Thread
P/4
P/8
Maximum Major Diameter of Internal Thread
Minimum Major Diameter of Internal Thread
Maximum Major Diameter of External Thread
Minimum Major Diameter of External Thread
Basic Major Diameter
Maximum Pitch Diameter of Internal Thread
Minimum Pitch Diameter of Internal Thread
Basic Pitch Diameter
Maximum Pitch Diameter of External Thread
Minimum Pitch Diameter of External Thread
Maximum Minor Diameter of External Thread
Minimum Minor Diameter of External Thread
Permissible Form of Thread from New Tool
Minimum Minor Diameter of Internal Thread
Maximum Minor Diameter of Internal Thread
EXTERNAL THREAD
(SCREW)

PLATE 2.—Tolerances, Allowance, and Crest Clearances for Classes 1A, 1B, 2A, and 2B.

Effect on Federal Standards.

The National Bureau of Standards publication dated November 1948, "Unified Screw Threads—As Recommended by the Interdepartmental Screw Thread Committee Representing the U. S. Departments of Army, Navy, Air Force, and Commerce", contains the unified standard adopted by international accord. The cover contains the following:

Tentative Draft
considering the
Report of Subcommittee No. 1
B1 Sectional Committee on the
Standardization and Unification
of Screw Threads(3)
and the Third Draft, Provisional
British Standard Unified Screw
Threads
CJ (MEE) 8309. UST 49(4)

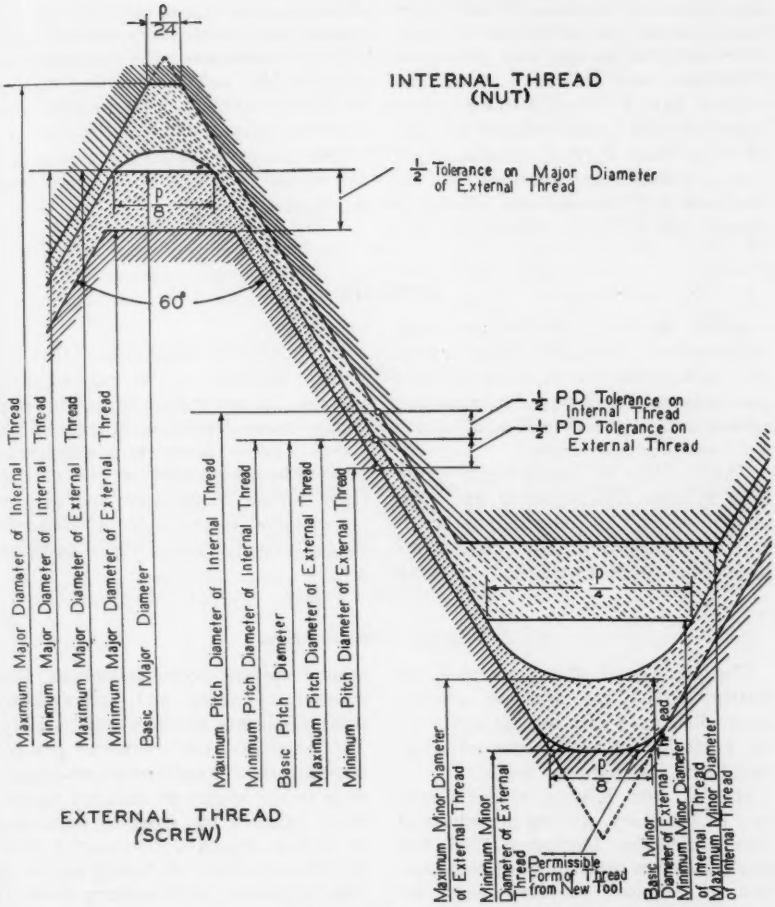

P/24
INTERNAL THREAD (NUT)
½ Tolerance on Major Diameter of External Thread
P/8
60°
½ PD Tolerance on Internal Thread
½ PD Tolerance on External Thread
P/4
Maximum Major Diameter of Internal Thread
Minimum Major Diameter of Internal Thread
Maximum Major Diameter of External Thread
Minimum Major Diameter of External Thread
Basic Major Diameter
Maximum Pitch Diameter of Internal Thread
Minimum Pitch Diameter of Internal Thread
Basic Pitch Diameter
Maximum Pitch Diameter of External Thread
Minimum Pitch Diameter of External Thread
EXTERNAL THREAD (SCREW)
Maximum Minor Diameter of External Thread
Minimum Minor Diameter of External Thread
Permissible Form of Thread from New Tool
Basic Minor Diameter of External Thread
Minimum Minor Diameter of Internal Thread
Maximum Minor Diameter of Internal Thread

PLATE 3.—Tolerances and Crest Clearances for Classes 3A and 3B.

Applicable quotation from the foreword of this publication is reprinted below:

The Interdepartmental Screw Thread Committee, established in 1939, consists of representatives of the Departments of the Army, Navy, Air Force, and Commerce and of liaison representatives of the American Standards Association. Its purpose is to promote uniformity in screw thread standards in the Departments concerned.

This Tentative Draft represents the standards for Unified screw threads in the form in which they will be submitted, for approval of the Departments, in the next revision of Handbook H28, Screw Thread Standards for Federal Services. In accordance with actions taken by the Committee, the mandatory portions of the Handbook will contain those classes which meet the requirements of the Departments, namely: Classes 1A and 1B

step tables for allowances and tolerances; classes 2A and 2B for all standard thread series and step tables for allowances and tolerances; and the existing class 3 for all series and step table until the Unified classes 3A and 3B have had further consideration. The non-mandatory portions of the handbook will contain the following: classes 1A and 1B coarse and fine thread series; classes 3A and 3B for coarse and fine thread series and step table for tolerances; the existing class 1 step table; and the existing class 2 for all thread series and the step table for tolerances.

The plates accompaning this article are taken from the above mentioned publication.

Application.

As the details are finalized and when Industry both here and abroad retool and start production to the new standard this work will be culminated and efforts ranging over some 30 years will bear profitable fruit.

There will be adjustments from time to time. No standard can long remain static. An international committee comprised of the various interests has been established to carry on the work.

It should be emphasized that no human device can be mandated to success. The success of any useful procedure must depend upon full understanding of its objectives and appreciation of its advantages on the part of those whose cooperation and support are essential to its proper development and to its operation as a useful function.

For Future Consideration.

The signing of this Agreement of Accord points the way to greater accomplishments and focuses again on the necessity for action both on a national and international scale.

Mr. W. John Kenney, Under Secretary of the Navy, writing in Industrial Standardization, December 1948, has emphasized the importance of coordination of Services' standards with Industry. Our war time experience has made us, both Industry and Government, as individuals and as Allied Nations, keenly aware of the importance of simplification and standardization to rapid industrial mobilization and its resulting ability to support military effort and therefore to the security of civilization.

The impact and strain of the recent war upon available resources has mandated accomplishment of continued simplification and standardization far beyond current accomplishments. This imposes a burden and responsibility both on military services and Industry. The services must wherever possible correlate their requirements and adapt them to the ability of industry to produce. Industry too must be persuaded to provide standard commercial items the characteristics of which are in as close alignment with military needs as possible.

After each war, new designs evolve requiring higher performance levels. Paralleling higher performance levels are the inevitable trends toward exhaustion of suitable continental resources. At the same time the trend of global conditions tend to confine the practicable use of such resources to those locally available. These conditions impose even more urgent demands for standardization among those whose purpose is common. They re-

quire optimum performance in the face of necessary conservation in all of its phases. Their solutions must enlist the best available talent and persuade the greatest effort to provide optimum coordination. Thus standardization should not be confined to local continental areas but should provide for international interchangeability on the broadest practicable scale.

The basis for unification and interchangeability accomplished by this Declaration of Accord was possible for one single and outstanding reason:

Administration, (from the sponsoring level of the Combined Production and Resources Board) down through the various functional levels, at all times, kept the matter of standardization at the working technical level of Industry and Government. Never once was the effort stultified by raising this level. If this situation can be maintained with similar freedom of action success of future similar undertakings is predicted.

REFERENCES.

(1) Combined Production and Resources Board Report on Conferences on Standardization of Screw Theads and Cylindrical Fits—London, August - September 1944.

(2) Combined Production and Resources Board Report on Conference on Unification, of Engineering Standards—Screw Threads, Pipe Threads, Limits and Fits, Drawing Practice and Metrology—Ottowa, Canada, September - October 1945.

(3) American Standards Association Report of Sub-Committee No. 1, B1 Sectional Committee on the Standardization and Unification of Screw Threads.

(4) British Standards Institution Third Draft, Provisional British Standard Unified Screw Threads CJ(MEE) 8309 UST 49.

NOTICE

The Annual Banquet of
The Society
in 1950
Previously announced for
21 April 1950

Has been scheduled now
to be held at
THE HOTEL STATLER
WASHINGTON, D. C.
on
Friday, 28 April 1950

FREE MACHINING STEELS—A report on Ductility and Impact Resistance.

BY W. LEE WILLIAMS.

THE AUTHOR.

MR. W. LEE WILLIAMS received the degree of Bachelor of Science in Chemical Engineering in 1941 from Antioch College. During his college career, four years were served under the cooperative plan as laboratorian in Physical Metallurgy at Battelle Memorial Institute. After graduation he became Metallurgist for the American Steel Abrasives Company. During the war he was commissioned in the Naval Reserve and served at the U.S. Naval Engineering Experiment Station, Annapolis, Md. After discharge, he remained as a civilian metallurgist and became Assistant to the Superintendent of the Metallurgical Laboratory. A previous article by Mr. Williams appeared in the November 1948 issue of the JOURNAL.

Introduction.

Free machining steels usually obtain their superior machinability through high sulfur or phosphorus contents. It has long been known that these steels should not be used for critical structures, especially where dangers of fatigue or impact are present, or where high stresses occur transverse to the rolling direction. Nevertheless, they are employed in large quantities where the cost of machining is an important consideration.

Nearly all free machining steels are produced in bar form. The great majority are turned into parts, such as bolts, in which the major stresses parallel the direction of rolling. In such cases the mechanical effects of the sulfur and phosphorus are expected to be at a minimum.

Only limited information has been published concerning the ability of free machining steels to deform and absorb energy prior to fracture. In two categories further research appeared warranted, and the Engineering Experiment Station undertook an evaluation of the following:

(1) The ductility of free machining steels stressed in the direction of rolling.

(2) The critical impact (embrittling) temperature when stressed in the direction of rolling.

Approach.

The experimental work was undertaken in two phases, one of which followed logically from the other. The first phase pertained to the study of nearly all types of commercially produced free machining steels, treated as a general class, to determine whether sulfur and phosphorus were among the predominant influences on ductility and impact resistance. The second phase consisted of a study of three series of steels in which all known variables were held constant while the sulfur contents were changed.

First Phase.

The steels tested in the first phase are listed in Table I according to increasing carbon content. They included basic open-hearth sulfurized carbon steels and acid Bessemer sulfurized steels, the latter, of course being high in phosphorus. In addition, several non-free machining basic open-hearth grades were included for comparison purposes. They were about equally divided between the hot rolled and cold finished conditions. Each steel was obtained as a single length of one inch diameter bar.

TABLE I.

Commercial Steels Under Test

Material Designation	Material Ordered		Actual Composition Percent					Nearest Actual AISI Number (1942)	Nearest Corresponding SAE Number (1941)	Nearest Corresponding SAE Number (1942)	Elements Varying from AISI Specification
	Composition	Finish	C	S	P	Mn	Si				
ABT	C-1020	HR	.10	.035	.055	.38	.01	C-1010	1010	1010	P .04 Max.
ABU	B-1113	CF	.12	.28	.083	.75	.02	B-1113	X-1112	1113	P .09–.13
ACO	B-1112	CF	.13	.174	.08	.81	.02	B-1112	1112	1112	P .09–.13
AEE	B-1113	CF	.13	.275	.096	.85	.03	B-1113	X-1112	1113	None
AEI	C-1010	HR	.13	.033	.011	.43	.03	C-1010	1010	1010	None
AED	B-1113	HR	.14	.25	.12	.83	.07	B-1113	X-1112	1113	C .08–.13
ACK	SAE 1112	HR	.15	.104	.03	.82	.03	C-1115	1115	1115	None
AEJ	C-1020	HR	.16	.033	.012	.49	.06	C-1015	1015	1015	None
ACM	C-1117	CF	.18	.16	.015	1.19	.01	C-1117	X-1314	1117	None
AEG	C-1122	CF	.18	.12	.011	1.46	.11	C-1122	.....	....	None
ACP	SAE 1020	CF	.20	.032	.007	.95	.21	C-1022	X-1020	1022	None
ACR	C-1117	HR	.20	.106	.010	.74	.16	C-1120	.....	....	None
AEF	C-1120	HR	.20	.12	.009	.81	.10	C-1120	.....	....	None
AEK	C-1020	CF	.22	.037	.005	.48	.13	C-1020	1020	1020	None
ACQ	C-1045	HR	.25	.039	.004	.74	.25	C-1025	1025	1025	Mn .30–.50
ACN	C-1137	HR	.38	.150	.010	1.61	.10	C-1137	X-1335	1137	S .08–.13
ACJ	Special	CF	.44	.109	.011	1.42	.06	C-1141	X-1340	1141	None
ACL	C-1040	CF	.45	.036	.008	.86	.15	C-1045	1045	1045	None
AEH	C-1141	CF	.46	.096	.010	1.36	.14	C-1141	X-1340	1141	C .37–.45
ACI	C-1144	CF	.48	.24	.015	1.58	.29	C-1144	.....	....	None
ACZ	AN Ni* Steel	HR	.32	.013	.008	.54	.20	A-2330	2330	2330	Mn .60–.80

*—Cr .083 Ni 3.38%

In a number of cases the composition was not exactly in accordance with the specification of the material which was ordered. Consequently, Table I shows the AISI designations which most nearly correspond to the actual compositions. Also included are lists of the SAE designations which correspond to those of the AISI. Finally, the column at the extreme right shows the elements which were at variance with the AISI specifications. In nearly all cases the variations were slight.

Each steel was tested in three conditions: as rolled, annealed and quenched and tempered at 1000°F. This tempering temperature was se-

lected so that all the steels, regardless of hardenability, could be machined readily after treatment. The quenching and annealing temperatures were selected according to carbon content, as follows:

Carbon	*Quenched*	*Annealed*
.10 - .20%	1650°F.	1625°F.
.21 - .30	1575	1540
.31 - .40	1550	1500
.41 - .50	1520	1450

Material ACZ, a Navy Class AN nickel steel included for comparison of impact properties, was tested in the following conditions:

(1) As received (hot rolled).
(2) Normalized from 1650°F., tempered at 1200°F.
(3) Annealed at 1450°F.

The tensile and hardness characteristics were determined for each steel in each condition. The individual values have not been reproduced here because they would add little to our understanding of the subject. Properly analyzed, however the data make order from confusion.

The treatment and analysis of the data presented some difficulty because of the variety of types of steels. The ideal way to study the effects of sulfur would be to hold all conditions constant except the sulfur content. In the present case, there were variations in all the other elements, including constituents not ordinarily determined. Furthermore, no control was possible concerning the foundry and mill

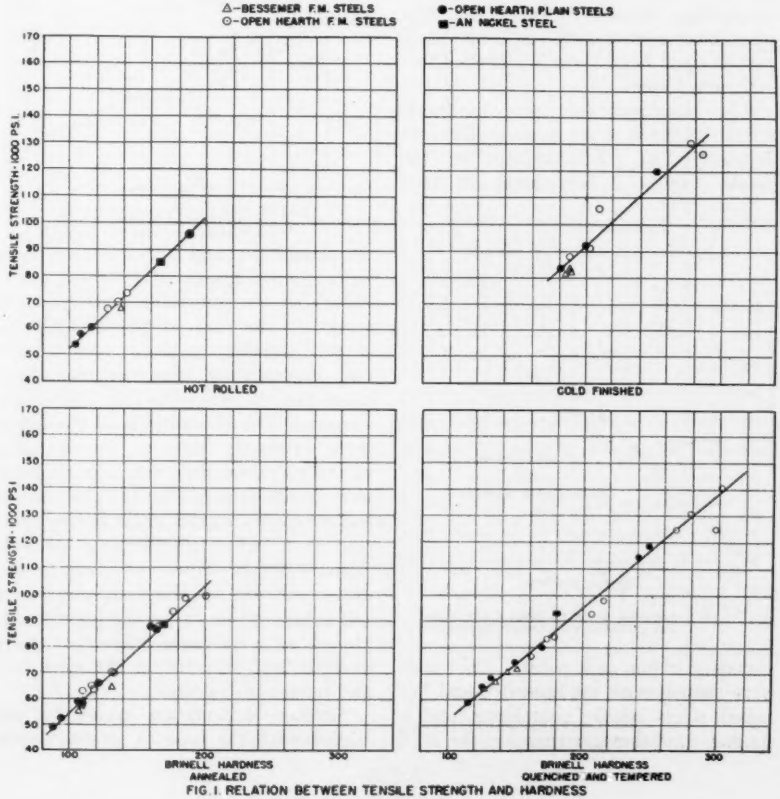


FIG. I. RELATION BETWEEN TENSILE STRENGTH AND HARDNESS

practices used in production. Hence, the best which could be expected of an analysis would be an indication of trends and general differences between plain and free machining steels as ordinarily produced by the steel industry.

It is common knowledge that steels, especially in the quenched and tempered condition, show a linear relationship between tensile strength and hardness. It is known further that sulfur and phosphorus have little effect on this relationship. In this respect the steels behaved as anticipated. Figure 1 shows the relation between hardness and tensile strength for each of the conditions of treatment. The individual points in the graphs, as well as in all subsequent graphs, were drawn with symbols differentiating between the free machining Bessemer steels, the free machining open hearth steels, the plain carbon open hearth steels and the Class AN nickel steel.

The locations of the hardness-strength curves were slightly different depending on the treatment of the steels. Figure 2 reproduces all four

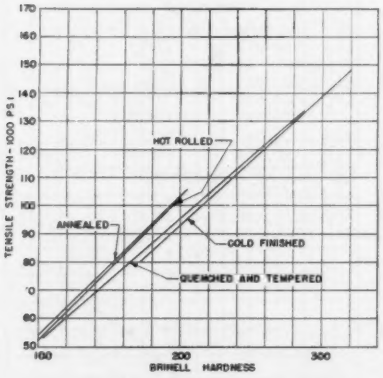


FIG 2 RELATION BETWEEN TENSILE STRENGTH AND HARDNESS FOR VARIOUS TREATMENTS

curves without test points. The lines were paired, with the annealed and hot rolled steels nearly coinciding, and a similar condition existing for the cold finished and quenched and tempered materials.

The plotting of data on ductility did not yield such precision, because variations in composition and other factors influenced the values to a considerable extent. One possible treatment of the ductility data is illustrated by Figure 3, in which the percentages

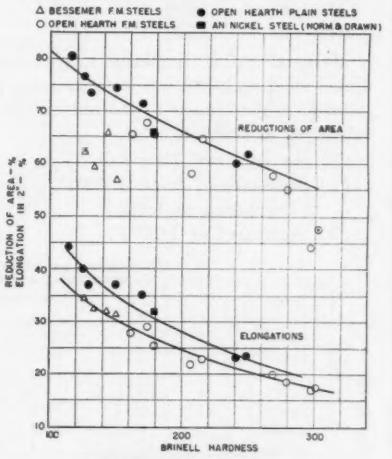

△ BESSEMER F.M.STEELS
○ OPEN HEARTH F.M. STEELS
● OPEN HEARTH PLAIN STEELS
■ AN NICKEL STEEL (NORM & DRAWN)
REDUCTIONS OF AREA
ELONGATIONS
REDUCTION OF AREA - % ELONGATION IN 2" - %
BRINELL HARDNESS

FIG. 3. RELATION OF BRINELL HARDNESS TO ELONGATION AND REDUCTION OF AREA FOR QUENCHED AND TEMPERED STEELS

of elongation and reduction of area were plotted against the Brinell hardnesses of the quenched and tempered steels. In the case of the elongations, it was possible to draw two curves, one representing non-free machining steels and the other the free machining varieties. The elongations of the latter were distinctly inferior. In the case of the reductions of area, a fairly good curve could be drawn for the plain steels, but the scatter was too great for the location of a curve representing the sulfurized steels. However, despite the scatter, it was evident that the points fell below the curve for the plain steels. Hence, as a class, the sulfurized steels showed less ductility in the quenched and tempered condition.

Similar information could be obtained with the type of graph shown

in Figures 4 and 5. This type of graph was preferred, and was chosen for the principal presentation of data, because it related the ductility to the carbon content. Hence, it presented a clearer picture of the relationship of ductility to the composition range of AISI grades of steel under investigation. The ordinate scale showed the ratio of the percent elongation to the Brinell hardness (or the reduction of area to hardness, as the case may be). Obviously, if two steels had the same carbon content and hardness, but different elongations or reductions of area, the one with the superior ductility would have the higher ordinate value. Likewise, since it has been shown that the

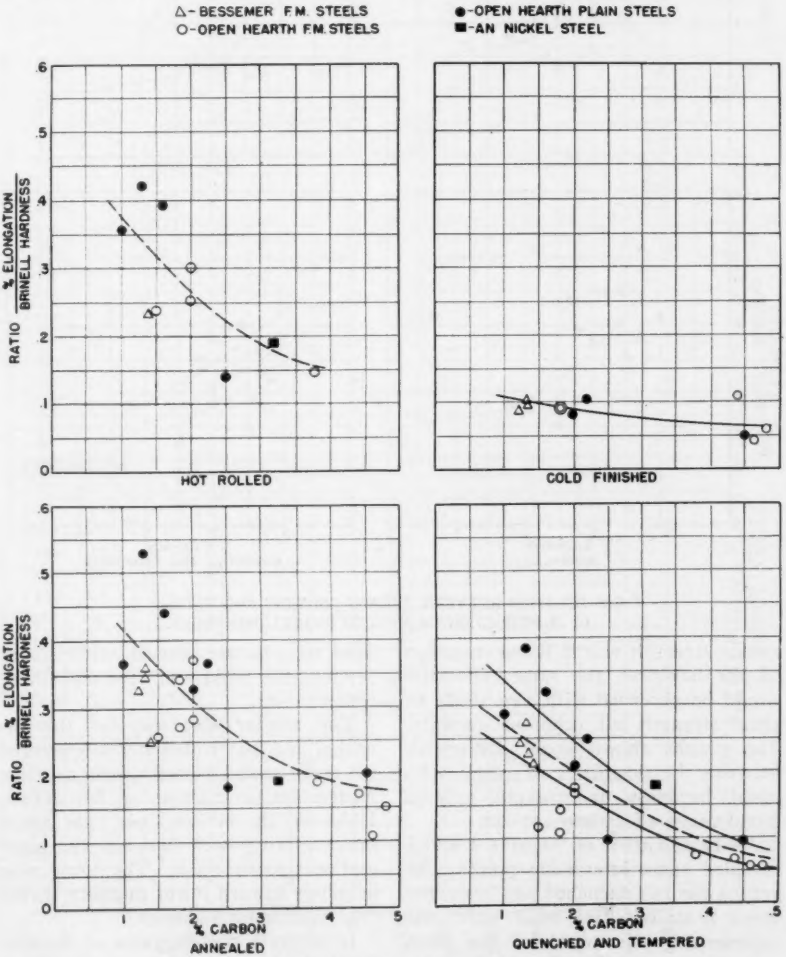


FIG 4 RELATION BETWEEN CARBON CONTENT AND RATIO OF % ELONGATION TO BRINELL HARDNESS.

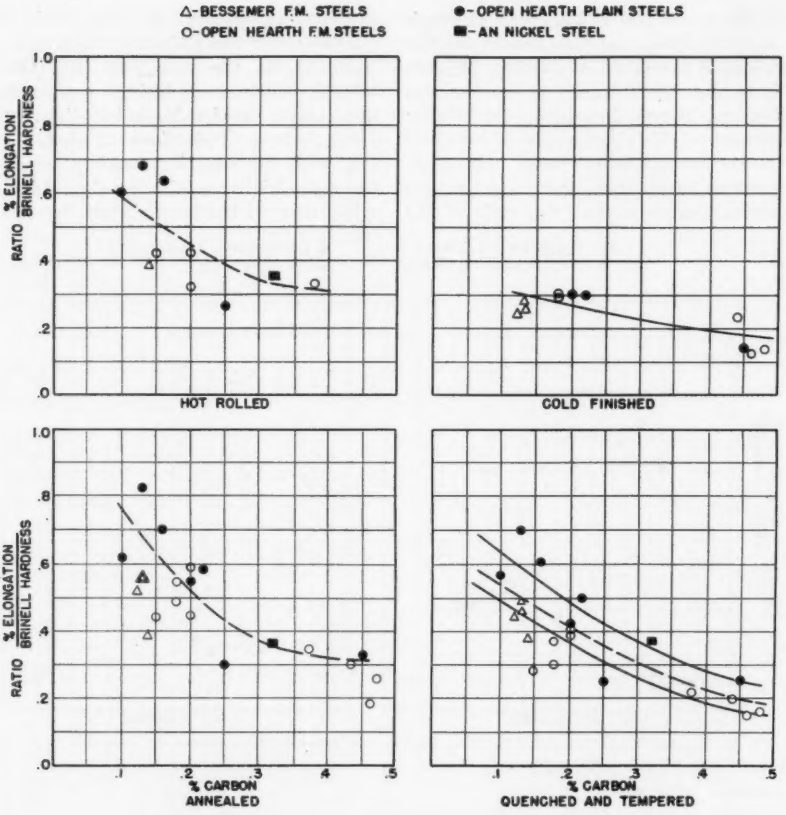


FIG. 5. RELATION BETWEEN CARBON CONTENT AND RATIO OF % REDUCTION OF AREA TO BRINELL HARDNESS.

tensile strength was a linear function of the hardness, the same difference would be observed with two steels of equal strength but different ductility. The graphs shown could distinguish between the ductilities of steels with equal hardness or strength over a broad range of carbon content.

An examination of Figures 4 and 5 revealed some interesting points. The graphs for the quenched and tempered steels contained two solid lines, one representing the trend of the plain carbon steels and the other of the free-machining steels. The latter were distinctly inferior with respect to ductility. It was possible to draw the broken lines to separate, almost without exception, the plain and free machining compositions.

The similar diagrams for the annealed and hot rolled steels presented too much scatter of test points to allow representative curves to be drawn. However, the broken lines once again divided fairly well between the plain and sulfurized steels. The trend was definitely toward lower ductility in the free machining varieties.

In contrast, the diagrams of the cold finished steels showed no significant difference between the two general classes of materials.

Standard Charpy keyhole impact tests of these steels were made at temperatures ranging down to—260°F. The object of the tests was to determine primarily the critical impact temperature, i.e., the temperature at which there is a change from a ductile to a brittle type fracture and a simultaneous marked drop in the energy required to rupture the specimen. In most cases the embrittling temperature was quite clearly defined. In others, the critical temperature covered a range of 30° to 40°F. In such

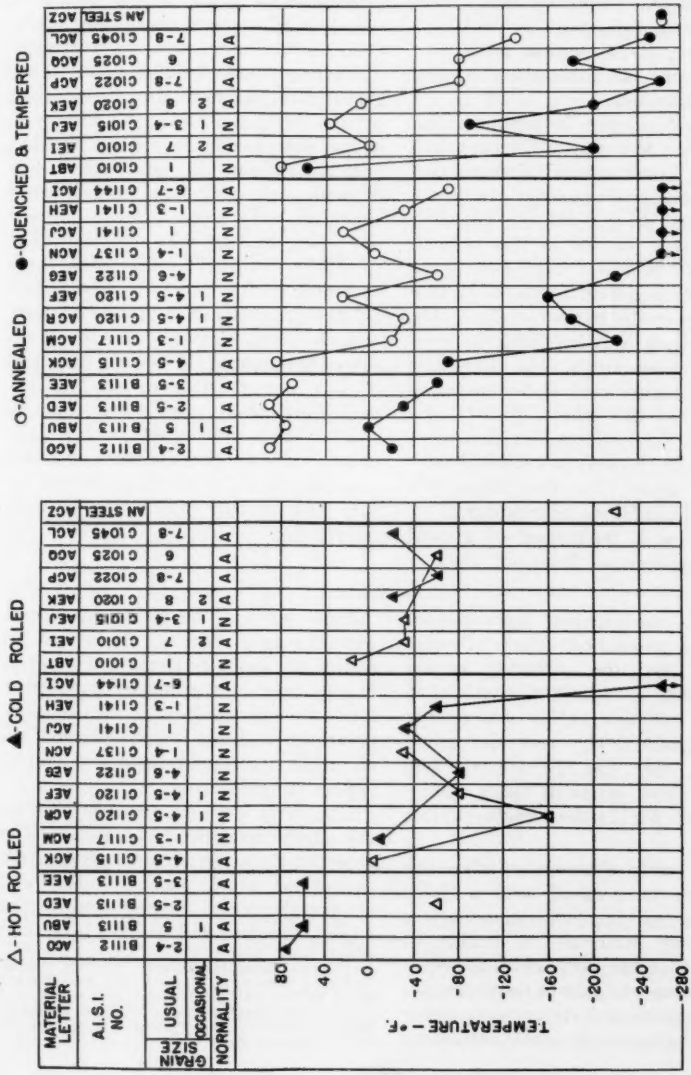


FIG. 6. CRITICAL TEMPERATURES FOR CHARPY KEYHOLE IMPACT TESTS.

cases the critical temperature was recorded at about the midpoint; such a selection was ordinarily associated with about a half ductile-half brittle appearance on the fractured surfaces of the specimens.

Figure 6 presents a diagram which summarizes the impact data. Each material was designated by its AISI number and its austenitic grain size and normality as determined by the McQuaid-Ehn test.

A review of the impact data indicated only a few trends. The quenched steels had lower critical temperatures than the same steels in other conditions of treatment. In general, the annealed condition was poorest in this respect, although a few minor exceptions were noted. The open hearth grades of free machining steels compared favorably with the open hearth grades of non-free machining steels. The Bessemer steels had higher critical temperatures than any of the others. Finally, the known beneficial effects of nickel were confirmed by the very low critical temperatures found for the Class AN steel in all conditions of treatment.

A very wide fluctuation of critical impact temperature, from one steel to another, was apparent in these diagrams. Such behavior has been noted by many investigators, and attempts at an explanation have been only partially successful. One of the known variables affecting the test is grain size. In the present case, all materials had actual grain sizes of 5 to 7, which eliminated that possibility as a significant variable. On the other hand, considerable differences existed in the austenitic grain sizes and normality as obtained by the McQuaid-Ehn test. However, it was not possible to explain the impact data from this standpoint. For example, except in the cold finished condition, very similar impact properties were obtained with material AEH (C-1141), normal and with large austenitic grains, and for material ACI (C-1144), abnormal and fine grained.

Such distracting circumstances occurred throughout the impact data, and it was not possible to explain the variations from one composition to another with these steels. Other variables, such as the foundry and mill practices, would have to be known before these impact tests could be fully interpreted.

Second Phase.

Whereas the previous tests demonstrated the pronounced effects of many variables on the critical impact temperature, it was not possible to determine whether sulfur was among those variables. It was fortunate, therefore, that the Laboratory had available three series of steels admirably suited for further study of this aspect.

In the early nineteen-twenties, the Laboratory cooperated with a Joint Committee of the American Society for Testing Materials to study the effects of sulfur and phosphorus in steels. During one part of the investigation, comprehensive tests were made on three groups of steel containing "added" sulfur. The published reports (1) (2) fully covered the effects of sulfur on many properties. However, in the case of the impact properties, tests were made only at normal temperatures with no effort directed toward the determination of the critical impact temperatures.

The steels were produced in 1920 by the Bethlehem Steel Company. The analyses were intended to cover three groups of steel, namely, plate and structural steels, forging steels and wheel, tire and rail steels. Each was

(1) Joint Committee Report, "Effect of 'Added' Sulfur on Structural, Forging and Rail Steels", Proc. ASTM, Vol. 23, Part I, 1923, p. 105.
(2) Joint Committee Report, "Final Report on Effect of Sulfur in Steel", Proc. ASTM, Vol. 36, Part I, 1936, p. 88.

made as a 45 ton oil-fired basic open hearth heat, with regular commercial practice followed as nearly as possible. Each of the three heats was poured into eight ingots, and sulfur, in the form of iron sulfide, was added during the pouring of the ingots. The ingots were allowed to cool, but not to atmospheric temperature, and these were reheated and rolled into 4″ by 4″ billets. These billets were bloomed by regular commercial practice. Bars 1″ diameter were rolled from one of the billets in the lower half of each ingot. These 1″ bars were those tested and reported herein. A more complete history of the steels will be found in the references.

The compositions of the ingots were determined as averages of 14 to 16 determinations. Table II contains the results. The group and ingot numbers correspond to the designations used in the ASTM reports and are used as a means of identity in the present report. All of these steels were normal and had an austenitic grain size of 5.

TABLE II.

Compositions and Critical Impact Temperatures of ASMT Steels.

Group No.	Ingot No.	Composition in Percent							Critical Temperature—°F.		
		S	C	Mn	P	Si	Ni	Cr	As Rolled	Normalized	Quenched & Tempered
II	1	.033	.20	.50	.013	.19	.19	.04			
	2	.039									
	3	.059							−40	−40	−150 (1)
	4	.066							−60	−15	−170 (1)
	5	.078							−30	−25	−155 (1)
	6	.074							−45	−40	−160 (1)
	7	.133							−45	−40	−160 (1)
	8	.136							−60	−40	−175 (1)
III	1	.026	.37	.60	.010	.21	.30	.04			
	2	.043									
	3	.060							−40	−40	−45
	4	.067							−60	−50	−50
	5	.080							−55	−50	−55
	6	.102							−50	−50	−50
	7	.113							−55	−60	−80
	8	.152							−65	−70	−80
IV	1	.033	.78	.63	.015	.20	.16	.05			
	2	.052									
	3	.060							120	100	80
	4	.071							100	100	90
	5	.083							100	80	100
	6	.113							100	100	100
	7	.128							100	120	100
	8	.153							100	70	90

(1) Quenched only.

The bars representing Ingots 1 and 2 from each group could not be located after so many years. Consequently, it was possible to test the steels only with sulfur contents of 0.06% and above. This limitation was not expected to detract from the usefulness of the test results.

The steels were tested in three conditions of heat treatment. For the sake of uniformity, the treatments were identical to those employed by the ASTM.

The material from Group II was tested in the natural, normalized and quenched conditions; Groups III and IV were examined in the natural, normalized and quenched and tempered conditions.

For Group II steels, the normalized condition was obtained by heating to 1600°F., holding 1 hour and cooling in air. The quenched condition was obtained by heating to the same temperature, holding 1 hour and quenching in water. For Groups III and IV, having higher carbon contents, the normalizing was accomplished by heating to 1500°F., holding for 1 hour and air cooling. The quenched and tempered condition was obtained by heating to 1500°F., holding 1 hour, quenching in oil, reheating to 1070°F., holding 1 hour and furnace cooling.

Standard Charpy keyhole impact tests were made over a range of temperature as described before. Figure 7 illustrates typical results as obtained with the Group II steels. The estimated midpoint of the critical temperature range was indicated in each case by an arrow. The location of the arrow was judged not only by the curve itself but also by the point at which half ductile-half brittle fractures were observed on the test specimens themselves.

In most cases the extremes in critical impact temperature were within the limits of experimental accuracy. The columns to the right of Table II summarize the critical temperatures found in all cases. No correlation could be observed between added sulfur content and critical temperature.

Conclusions.

Realizing that wide differences can exist between two lots of the same type of steel, or betwen steels of different composition, the following conclusions appear warranted in considering sulfurized steels as a class when compared to plain carbon steels as another class. All conclusions are based on tests in which specimens were cut and tested parallel to the direction of rolling.

Within the range of hardness usually encountered, there is no significant difference in tensile strength, for equal hardness, between free machining and plain carbon steels.

In general, free machining steels as a class have slightly lower ductilities than corresponding non-free machining varieties of equal hardness for the hot rolled, annealed and quenched and tempered conditions. However, in the case of the steels, investigated, no such distinction is noticable for the cold rolled condition. The differences in ductility are reflected by elongation and reduction of area measurements.

For any particular steel, including the plain and sulfurized varieties, the quenched and tempered condition results in the lowest critical impact temperature as determined by the Charpy keyhole test. Usually the annealed condition gives the highest critical temperature.

The basic open hearth grades of free machining steels compare favorably with the basic open hearth plain carbon steels with regard to critical impact temperature. The acid Bessemer steels are inferior to the open hearth steels in this test. The addition of nickel, as used in material ACZ, lowers the embrittling temperature very effectively.

No correlation can be observed between added sulfur content and critical impact temperature. The effect of sulfur in this respect, if present at all, is insignificant to the other variables known to have pronounced effects on impact behavior.

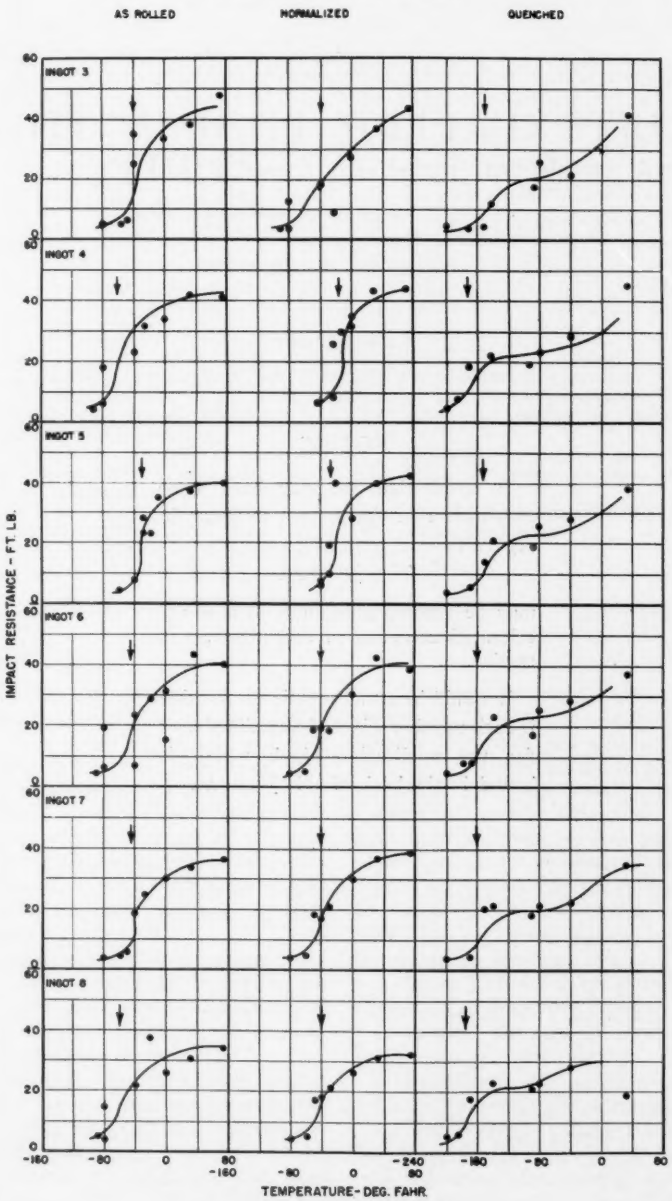


FIG. 7. CHARPY IMPACT TEST RESULTS ON HIGH SULFUR GROUP II STEELS.

NOTICE

In order that we may have some assurance that the present program in regard to

The JOURNAL

may be accomplished, a somewhat accelerated accretion of new members is needed.

It would be an automatic proposition if each member will find a qualified user for the application blank which is bound in the back of this JOURNAL.

LIGHT WEIGHT WELDED STRUCTURES—A challenge to Naval Architects in Ship Design.

BY SAMUEL WEENING.

THE AUTHOR.

Born 1896 into a family of Salt Water men and shipbuilders—Brooklyn, N. Y.
Educated in New York City, graduated from Townsend Harris Hall (C.C. N.Y.)—furthered education, Brooklyn Polytechnic Institute in Thermodynamics and Propulsion.
Served apprenticeship in Robins Drydock and Repair Co.—worked as draftsman, design engineer, hull superintendent and Senior Estimator—24 years with Federal Shipbuilding and Drydock Co.
Wrote a history of American Shipbuilding for the Norwegian Study group and designed a "two level" yard for the Norwegian government.
Now specializing as an industrial consultant.
Obtained patent #2297040 entitled "Structural Units".

FOREWORD.

Reduction in weight and cost of metallic structures is receiving increasingly more attention on land and sea; the many technical magazines and 'papers' presented before technical societies demonstrate the trend of such attention. While the scope of attention so devoted by engineers is ample, the quality and nature of the same is subject to improvement and generally is so conceded by engineers.

Metallurgists and welding engineers have directed much of their attention to the often obscure causes of structural failure. Between them they have made countless studies of 'notching'.

Inasmuch as medium carbon steel is still the favorite structural material it follows that the bulk of research data is found in that field.

Steel trade organizations and steel producers themselves have joined the host of engineers and other specialists in exploring the vast fields open to structural research. So large is the amount of current available data that no one person could hope to read and much less, study any more than a small amount of such information. There have been however, reviews and symposia serving to concentrate data as well as serving to strip such data down to the pertinent and applicable. One could list the foregoing fields under the following headings, viz;

1. Stress Concentration through Inadequate and Improper Design.
2. Failure through Overstressing, Progressively and Critically.
3. Refinement of existing design methods, orthodox and Heterodox.
4. Metallurgical and Mechanical Notching, Fatigue, Creep and Growth.
5. Vibration, Dampening and Absorption in Dynamically and Critically Loaded Structures.

There has been surprisingly little advancement made in land and sea structures when such are compared with modern heavier than air craft. The structural efficiency of heavier than air craft is very close to the ideal, yet aeronautical engineers unceasingly strive toward improvement in *all* lines to produce a lighter and stronger wing, a larger and lighter fuselage, etc.

Doubtless, the complete lack of arbitrarily established technical data at the incept of air craft construction imposed the need of a completely new start in accurately predicting and meeting the requirements of an utterly new and complicated mechanical device.

It may be safely stated that the aeronautical engineers and scientists have advanced further in three or four decades than their more static brethren have in two centuries. Those responsible for the design of air craft did not allow themselves, in the final stages, to be influenced by arbitrary rules or empirical formulae but assigned themselves the duty of thoroughly proving out practically, that which had been planned mathematically. Step by step the ever increasing demands of rising horsepower and useful load carrying capacity have been met with great skill. The degree to which their science has advanced has in some cases become a pattern followed to a small degree by structural engineers.

There are many external influences such as antiquated Codes, etc. which heavily saddle Marine and Structural engineers with out-dated and laborious procedure, hampering directives and unnecessary so called 'protective features'. These things are all very well known but like the mice in conclave "Who will hang the bell on the cat"? Further, an 'old saw' comes to mind such as not 'Seeing the trees for the woods', when the whole panorama of present structural research is viewed impartially.

Referring to the headings or titles for research (items 1. to 5. inclusive) it does seem that items one, two and three are fields that need reexploration. Fields four and five have received the preponderance of truly scientific attention, although often repetitious; Yet, fields four and five have not aroused the intellectual curiosity in the fields one, two, and three equalling the restless scientific pursuit in air craft development.

It will be subsequently shown that much lighter and more efficient structures can be achieved on land and sea by the judicious application of basic information gleaned in fields four and five. The method of such demonstration is somewhat unique and therefore subject to adverse criticism. That is as it should be, and is part of the scheme of things!

The increased knowledge of stress flow in structures must, as time goes on, point the way to a more efficient application of structural materials whether steel, wood plastics, aluminum or other media, for these are all heavily subject to the inexorable laws of stress flow. It is with that picture in mind that the subject of Light Weight Structures is set forth herein.

There is much still to be done in the field of light weight structures and it is hoped that the matter following herein will serve to stimulate further investigation and improvement by those better qualified than the author. Detailed calculations and supporting data have been held to a stark minimum for the avowed purpose of stimulating further research if only through animal curiosity.

When steel making advances from the status of the arts into the status of the sciences a new field of design will be opened whose vista is still too distant to be viewed. Remember, we still make steel like 'concrete' and subsequently subject it to various treatments to make it behave less and less like concrete! We hope that the steel makers continue to excite our attention sufficiently to goad us into ever striving for improvement in design and application.

History and Underlying Present Tendency.

Primitive man, whether he lived in caves or clung precariously to trees had scant knowledge of cause and effect in nature about him. His functioning as a fairly rational animal was limited to the demands of merely staying alive.

When in the course of his migrations the need arose for creating his own shelter in place of moving into a natural one, man's mental faculties were pushed into activity by such need. It may be safely stated that his ability to cope with problems arising out of creating his own shelter has always been restricted to some extent by inadequacy. Even now, much as we know of structural materials and their application we are still very much subject to the whims of the unpredictable group of natural forces such as earth quakes, tornadoes, isolated waves of unusual height, etc.

The early use of stone in structures was not likely due to selectivity or by design but rather due to the sufficiency of usable stone at hand and the virtual absence of cellulose material.

We may flatter ourselves that we do not choose the path of least resistance in our efforts to maintain creature comforts; be that as it may, we exert ourselves no further than the instant the *first* pleasing solution presents itself. The exception of course is to be found in efforts exerted by scientists or others trained along specific lines.

Unflattering as it may seem, there are times when we, all of us, find ourselves reacting to something akin to instinct—possibly race canniness. Our efforts as practicing engineers too often and too easily conform only to a single line of procedure without making any rational attempt to coordinate it with a necessarily complementary line of procedure. It is further unlikely in our rather easy course to resort to thorough integration of several lines of procedure. It has led us into some fantastic inconsistencies.

Here, we surely have evolved a controversial issue, an issue however having some foundation.

Rhetorical questions are at times classically permissable to politicians and not often to engineers, but a few should be set forth to stir our imagination.

In building our dwellings, ships, and in many diversified structures do we truly and painstakingly select each part for its proper load carrying capacity or in a manner to resist stresses with a minimum contribution of weight?

In our houses do we not incur mechanical notching in effect by the juncture of joists and studs at right angles, and on ships do we not virtually create the same effect? For the defense of our ship procedure one may answer that brackets are applied. The question still holds however, in that the bracket also invites mechanical notching if it is not swept up into a proper curve and in such a manner that along the major axis of stress the geometric section is not bi-symetric. It may be said that the cost would be intolerable. Doubtless that is true, but why let it rest there? Why not throw off our self imposed restrictions and make a serious attempt to discover just *what does make things tick?*—and when they 'tick' they should 'tick' in harmony.

We have not done with the Rhetorical gripes just yet! For a moment let us pause at the cargo vessel. Why the rectangular cargo hatch with its crude accompaniment of adjacent doublers placed by more or less scientific 'guess' and often its very existence an accident going some place to happen! Why the jungle of cargo handling gear with its maze of running and standing gear inviting wind and sea to 'do its stuff'?

Why on certain light combat vessels have we permitted (of all things) the proximity of aluminum and light steel?

What goes into making such fire works as a thermit weld?

Why a stainless steel door for a combat vessel when the vessels tactical qualities are long since obsolete when *that* door is still like new!

These Rhetorical questions dealt with secondary structures in part, now then, can we still smugly presume somehow our primary structures are just right? That, should at least provoke some kind of an answer. Actually we all feel that our primary structures are not just right but they are what we have been permitted to produce!

Formally acceptable methods of inductive and deductive reasoning have undergone considerable change and more than considerable improvement in two centuries (that time spanning the years of the mechanical prime mover with all its statical progeny). Some methods have long ago been abandoned and justly so, and some methods although known into the dim past have not even been noticed in recent times. Lastly, just to demonstrate what is meant, consider this one—We have it constantly dinned into our ears and minds that it took eons of time to make our coal beds—if that is so, 'how come' the phosphorus did not go back into the atmosphere, it being one of our most volatile substances? Further how is it that no one sees anything significant in the fact that most of North American coal beds within the so called last ice age belt contain countless thousands of 'petrified' trees *broken at the bole* and lying with the trunks in a generally Southerly direction? So apparently we even question some of our reasoning methods and we introduce our subject with a rarely used one.

INTRODUCTION.

In past years the writer has found analogies a very helpful expedient in coaching aspirants to our Naval and Military Academies. At times some analogies are of a nature which tends to withold complete enlightenment for a while until a certain pattern of concepts may be established.

Living matter furnishes us plentifully with useful analogies, in fact, nature often sets before us a panorama of biological and mathematical cause and effect wherein one can perceive in a short time a contraction of what otherwise is cycle after cycle material development. For example, the animal foetus from conception to birth is a quick preview of animal's rise from a single cell creature to its present state. It is of interest that the foetal newt, dog, cat, shark, human being, and monkey are strikingly similar in the early stage of foetal development.

To approach our subject more closely; nature actually shows us how and why its living and growing beams and columns withstand the external forces but is somewhat secretive when it comes to divulging the 'whys' and 'hows' of resistance to internal forces. Inasmuch as the metallurgists and welding engineers have already done a good job on internal forces for us this is fortunate and leaves us with the problem of resistance to external forces.

It is well at this point to bear the following in mind;

1. Certain trees (as cantilever beams) can stand isolated where other species so isolated would go down in the first violent wind. The latter must grow closely together. Their construction from top to roots are geometrically different from the former.

2. Certain grasses and reeds get their resistance to external forces by their own construction plus the close association to each other.

3. It has been often demonstrated in metallic crystal study, that the cumulative resistance of closely associated crystals is far greater than the unit

resistance of a single crystal would indicate.

Inasmuch as the core of this article is so thoroughly framed with an analogy of growing things, the following statement is pertinent though still a subject of controversy.

Man did not create or invent the mathematics, he discovered them as he learned the secrets of matter and matter in motion. Why? Is it not fundamentally true that the volume of a cube is the cube of one linear dimension and that the volume of a sphere is $1/6\,\pi\,D^3$ whether man exists on this earth or not? By the same concept the same is true for the calculus and the vast field of the Theory of Numbers. Thus if nature expresses its laws of matter by the mathematics, then surely its own creations must automatically meet such laws, and when nature resorts to hazards we have animal or cosmic catastrophes. So, even nature must obey its own laws!

The Light Structural Unit.

Having set forth our problem in a less than orthodox fashion, we may now move into the examination of the current situation.

The manufacture of steel and its various chemical compositions as well as physical properties have all been given attention and study enough to meet the practical demands of the moment. That is all that can be reasonably expected of those in the steel business, but the time is rapidly approaching when the engineers must apply steel products more efficiently!

If we must still look forward to the stark possibility of war, then whatever we entertain for its successful (to our arms) conclusion must definitely be designed with the maximum efficiency, dollars input to dollars output! 'Tailor made' items should be discouraged where possible, and devices that can be made by a continuous process encouraged. This must be done, for when tax saturation has been reached and industry cannot be paid out of real funds, we automatically reach for statism. We can avoid that!

A light Structural Unit must be produceable on a continuous basis and be universally applicable.

It is proposed herein that steel as we can now receive it is satisfactory, and that it is up to us to use it correctly with masses properly disposed, and in *paths* serving to *distribute,* rather than to absorb by mass, the stresses imposed from without; further that the geometric patterning is such that the units of stress per units of section are so minimized, that absorption could not be of import.

Structures in failing under stress, fail critically or progressively, or both, in part. The elimination of critical stress, and the design for progressive failure is the desired end. Not all authorities are in agreement in defining critical stress. The author believes the following differentiation reasonable.

A column fails by critical stress when during loading it bends in only one or two places with the remainder of the column in good condition locally, with its use as a column no longer available. A properly designed column under similar loading would gradually and uniformly wrinkle progressively, (like an accordian). See Figure 1.

A solid column can be made to fail critically, and a hollow column similarly, but a solid column *cannot be made to fail* by wrinkling. It behooves us therefore to examine the solid column before proceeding further.

To have a solid column fail progressively and have all its elements ultimately stressed, some phenomenon other than wrinkling must occur.

1. That column must be short enough

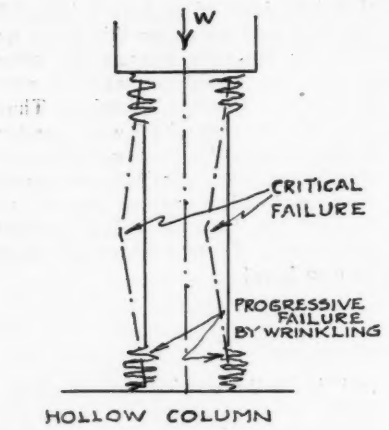


FIG. 1

CRITICAL AND PROGRESSIVE FAILURE

in length, relative to its diameter so that upon overload in the direction of the axis, all particles of the column tend to move *away* from the axis.

2. Assuming that the solid column is roughly made up of an infinite number of smaller columns held together by some acceptable affinity, each little column *wants* to bend critically but cannot do so, in that a neighboring little 'column' also wanting to fail critically, does not 'choose' to fail in the same direction.

3. Something must happen, and it does, See Figure 2. All particles will move. The 'row' of particles (little column) in the exact center of the column gets shorter along its axis and must therefore gradually become 'fatter'* with its largest diameter ocurring midway between the ends of the column in uniform material.

The next adjacent row of particles ('little columns') surrounding the 'little' column at the exact center will definitely be deflected away from the axis, having its direction imparted by the 'fattening' of the center 'column', and at a rate no faster than the transverse movement of the center column in its process of 'fattening'. (Molecular affinity in homogenious material) this row of columns also 'fattens' but to a *lesser* degree than the center 'column'.

4. Each successive annular row of little columns is gradually less 'fat', but bending more until the outer or 'skin' row is reached, which does *not 'fatten' but 'bends'*.

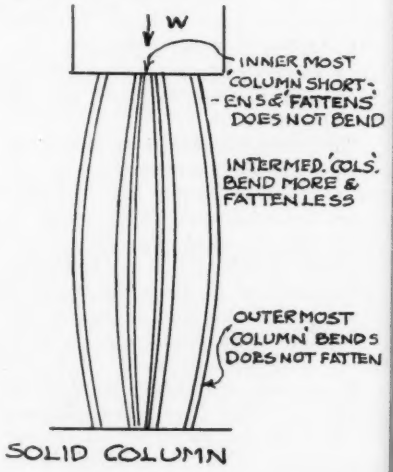

W
INNER MOST COLUMN SHORTENS & FATTENS DOES NOT BEND
INTERMED. COLS. BEND MORE & FATTEN LESS
OUTER MOST COLUMN BENDS DOES NOT FATTEN
SOLID COLUMN

FIG. 2

If the solid column were not truly homogeneous under the above conditions, or suffered from inclusions or piping, the column would split lengthwise if the linear affinity between particles were strong, and transverse affinity between rows weak, or the column would fracture at an angle or angles from the axis, if the linear affinity were weak, and the transverse affinity strong.

We eventually conclude that no matter how we proportion a solid column in steel it is a wasteful method of making a column!

* This phenomenon should *not* be confused with the small amount of volume loss due to compression, sometimes referred to as Poisson's Ratio.

A hollow column could be designed, considerably lighter, more efficient and carry the same load as a solid column of the same length. Such a column upon being overloaded would, or should fail as indicated in Figure 3.

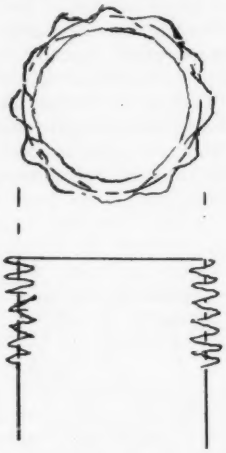


FIG. 3

PROGRESSIVE FAILURE PATTERN

In a hollow column properly designed, on the application of overload, the particles of the column gradually pattern themselves uniformly about a hypothetical cylinder somewhere between the outer and inner walls of the column. This pattern follows a progressive 'zig zag' course about the 'cylinder'. Such process is wrinkling, and being progressive. *should start* at the *ends* of the column. Now we have an important distinction. To keep a *solid column* from failing critically it must be designed to fatten, starting *midway* between the ends. Whereas to keep a *hollow* column from failing critically it must be designed to start wrinkling at the ends. The movements involved, their causes etc., in the aforementioned wrinkling process are omitted herein to avoid over-lengthy dissertation. Such information is available in any good text book on aeronautical structural design.

In early investigations made on the application of hollow columns, the path led to the necessity of proving or disproving the value of the analogy perceived in the arrangement of particles established by nature to resist external forces imposed on living matter. They were, the mammal's body, the deciduous tree and the grasses, particularly bamboo.

The search first came to rest on the elm tree which stands very well, when isolated.

The elm tapers concavely along the trunk with the slope increasing gently until the point of maximum moment is almost 'hard by', where the slope increase becomes more pronounced and more rapid into the roots. See Figure 4. Here we have in fact a column that is a cantilever beam.

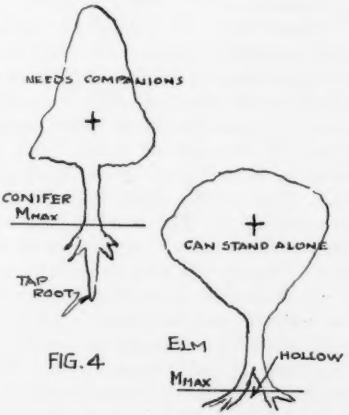


FIG. 4

Most trees, the elms included, have an area of pith or similar softness in the axis. Such pith section increases toward the bole, until in such trees as the elm, nature flares out the roots and causes the pith area to flare out into an area that is virtually hollow and *that is significant! Nature here has nicely provided against stress concentration.*

Stress Concentration.

Many scientific papers have been written on the subject of Stress Concentration in loaded structures and devices. The more recent and definitely enlightening ones have been illustrated by the photo elastic method of analysis.

The whole subject of stress concentration was hopelessly confused both as to theory propounded and as to professed proof set forth. The advent of the photo elastic method of analysis has cleared the atmosphere and become a very important and accurate tool in the actual measurement of the intensity of fibre stress and the patterning of the paths. We are indebted to M. M. Frocht and William M. Murray who have both written authoritatively on this very interesting subject. Karpov did much pioneer work in practical experiment with welding and riveting steel and his findings are substantially as follows;—

"After forces are applied to a structural member, stresses are developed at each point of it and as a general rule in three directions. The visualization of stress conditions may be obtained by the use of stress fields or stress lines analogous to the fields and lines used in the study of electrical phenomena . . . The flow of stress in many respects is very analogous to the flow of liquid. As long as no changes in conditions occur, the flow is more or less uniform and no stress concentrations occur. Any change in conditions disturbing the uniform and smooth flow results in Stress Concentrations . . . More abrupt changes in flow conditions are responsible for higher stress concentrations . . . The *boundary* will influence the stress conditions in the same way as the friction at the boundaries will influence the condition of the bounding layer of a flowing liquid."

Mr. W. M. Murray* in his article in "Metal Progress" of February 1941 Pages 195-200 gave a very good and lucid example of photo elastic analysis. Other papers of a similar nature and conclusions have since been published. A Typical example is illustrated in Figure 5. Essentially the various "papers" select a model cantilever beam of some material, (usually a plastic), lending itself readily to light refraction under stress of the model. The results of the model tests disagree sharply with simple beam formulae at approximately the point of maximum moment, as indicated in Figure 5.

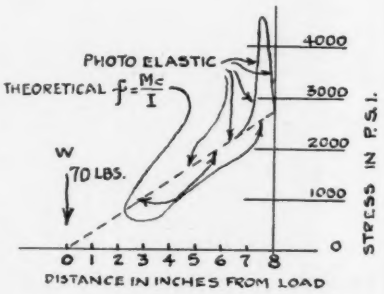


FIG. 5

COMPARISON: CLASSIC FIBRE STRESS DERIVATION WITH PHOTO ELASTIC OBSERVATION. NOTE DIFFERENCE AT POINT OF MAXIMUM MOMENT

In Mr. Murray's experiment, the maximum stress computed by the photo elastic method was 4350 lbs. P.S.I., while that determined by the beam formula was 2700 lbs. P.S.I. The ratio of these two values gave a stress concentration factor of 1.61 for the particular case.

Following the thought aroused by these experiments it would seem that the next logical step in dealing with the general condition is to

(a) Increase the Section Modulus in keeping with the photo elastic analysis.

(b) Remove a certain mass of properly determined area in the vicinity of

* Asst. Prof. Mech. Engineering M.I.T. Cambridge, Mass.

the neutral axis, the point of maximum movement, and the area of beam restraint.

The Elm does what is stated in (*a*) *and* (*b*).

Invariably where solid masses meet solid masses at sharp angles we have had it proven to us by photo elastic analysis that there is a 'bunching' up of stress pattern. That has been found consistently to be the result by various heavy industries, government bureaus and the U. S. Navy.

In the previous statements, stress flow has been likened to liquid flow, and yet that does not quite give us a clear picture of why we have heightening of stress values at points where forces have been piled up or concentrated, therefore, an attempt is made by Figure 6. to make the reasoning more acceptable.

Let us assume that in Figure 6 we have represented tunnels of flexible material, say, light steel, in which a mass of men moves continually and generally in one direction and at times at a rapid rate and sometimes running at top speed. For illustrative purpose, these men are to move into tunnels from other tunnels. In this instance

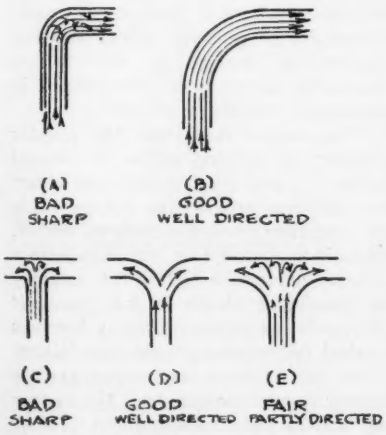


FIG. 6
STRESS DIRECTION & CONTROL

the expression 'Man Movement' is advised, because, in addition to moving concertedly in a specific direction, he swings his arms, bumps into his fellow, scrapes and bumps along tunnel sides etc.—'he vibrates' and so will the tunnels through which he moves!

On inspecting 'Man Movement' in Figure 6 it becomes obvious that such movement must be directed and gently confined to properly curved tunnels of sufficient section for the number of men passing a given section in a period of *time* that avoids 'bunching' up of men. These 'men' when forced to bunch up through excessive pressure behind them to speed up, will spend more time in bumping each other out of line of required direction than they spend in moving forward, also when the point of direction change is reached they find the change too abrupt we have the "fist fights" and tunnel sides "dented".

In Figure 7, five tunnels A, B, C, F, and G are loaded with men moving at a given speed to enter transverse tunnels X-Y to be directed into tunnels D and E. The men in these tunnels should move as smoothly as liquid or air flow. The positioning of tunnels F. and G. is such, that obviously tunnels D. And E cannot receive men from tunnels F and G without turmoil and the only other approach possible without turmoil is via tunnels above or below the plane of this page. This principle is violated considerably on shipboard to facilitate auxiliary services!

Returning to the Elm tree, it is analogous to a single column, but a single column study is not the purpose intended herein, and the Elm was selected for a stress analogy; however, the hollow bamboo comes in for a closer scrutiny.

It has been noticed that grasses and bamboo stalks render mutual assistance to each other in withstanding wind load and that there were several principles being 'previewed' thereby. Ob-

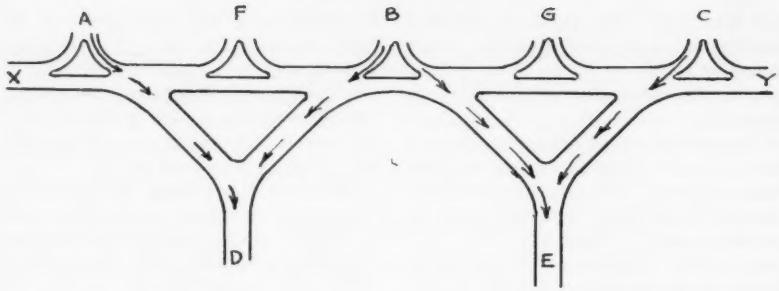


FIG 7.

STRESS ANALOGY; MASS MAN MOVEMENT AND TRANSFER TUNNEL TO TUNNEL
THE ANALOGY INCLUDES THE ASSUMPTION THAT RECEIVING TUNNELS ARE SUFFICIENTLY ENLARGED TO RECEIVE INCREASED VOLUME OF MEN TO MAINTAIN CONSTANT SPEED

viously the friction between the stalks plays an important part and is directly discernible. The bamboo stalk however has a high degree of resistance in itself, otherwise stated,—'structural integrity'.

Figure 8 shows the following characteristics of bamboo.

1. Long, extremely thin outer fibres. They are very tough as well as hard.

2. Under the microscope these fibres reveal small granules or blocks enmeshed.

3. Hard rings or annular 'lumps' occur periodically along the stalk at right angles to the axis. These, incidentally are the cicatrices or scars left by the blades or leaves.

4. An apparent diaphragm transversely to the axis and in the same plane as the cicatrix or 'ring'.

The apparent diaphragm is a residual spongy substance needed in the feeding and growing process but having no structural value. Careful removal of this spongy substance reveals a hard lump or ring in the same plane as the cicatrix.

When the hard inner ring is ground away the bamboo stalk fails critically under load as a beam or a column. But if the 'lump' is left intact the bamboo fails approximately progressively, with the adjacent longitudinal fibers pulling away from each other and the lump staying *intact!*

Were a column made similar to a bamboo stalk but in medium carbon steel the longitudinal 'fibres' would proportionately have a greater affinity for each other, and accordingly—'go nature one better', because we could design for progressive failure between diaphragms where as the failure occurring between the diaphragms in bamboo is essentially critical.

That means then, that the greater number of failure points or flexual points for any given length, the closer one achieves progressive failure. As is so often the case in the study of matter, organic and inorganic, we deal with a relative perfection and in this analogy in particular there is no absolute! There is no sharp division between critical failure and progressive failure. Note this. If these diaphragms are *too closely spaced we approach the danger of critical failure once more.* If they were spaced close enough we would

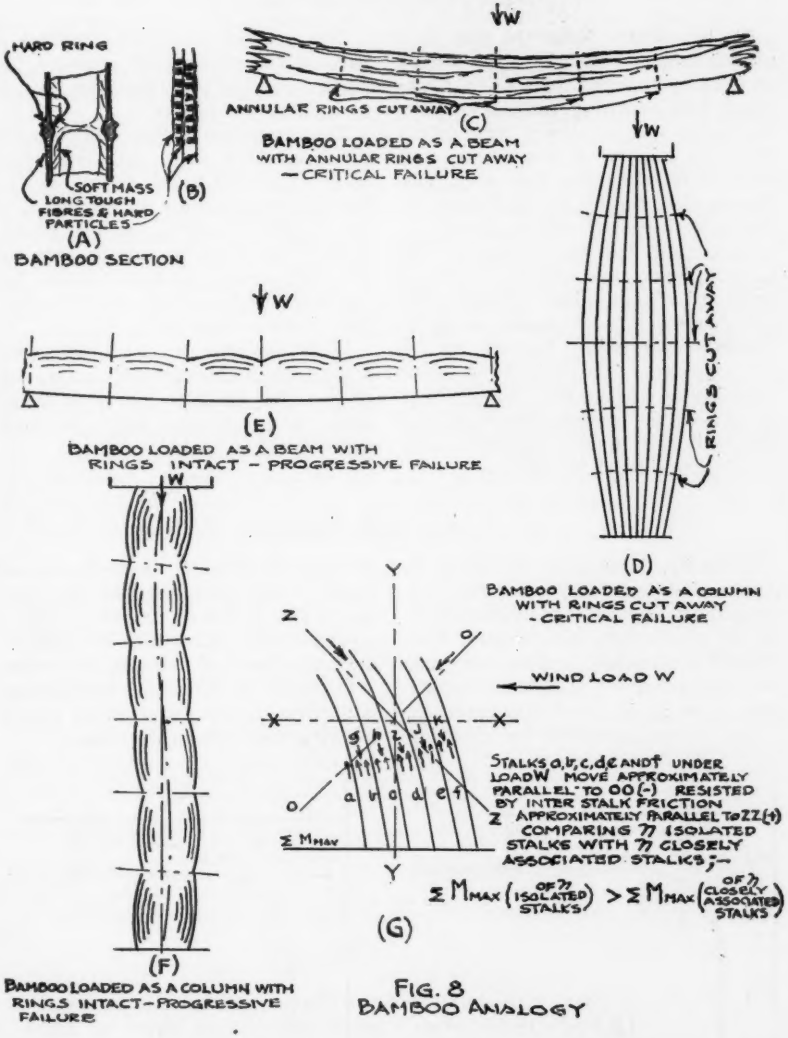

HARD RING
SOFT MASS
LONG TOUGH FIBRES & HARD PARTICLES
(A)
BAMBOO SECTION
(B)
W
ANNULAR RINGS CUT AWAY
(C)
BAMBOO LOADED AS A BEAM WITH ANNULAR RINGS CUT AWAY — CRITICAL FAILURE
W
RINGS CUT AWAY
(D)
BAMBOO LOADED AS A COLUMN WITH RINGS CUT AWAY - CRITICAL FAILURE
W
(E)
BAMBOO LOADED AS A BEAM WITH RINGS INTACT - PROGRESSIVE FAILURE
W
(F)
BAMBOO LOADED AS A COLUMN WITH RINGS INTACT - PROGRESSIVE FAILURE
Y
Z
O
WIND LOAD W
X
X
Y
STALKS a, b, c, d, e AND f UNDER LOAD W MOVE APPROXIMATELY PARALLEL TO OO(-) RESISTED BY INTER STALK FRICTION Z APPROXIMATELY PARALLEL TO ZZ(+) COMPARING n ISOLATED STALKS WITH n CLOSELY ASSOCIATED STALKS;-
Σ MMAX (OF n ISOLATED STALKS) > Σ MMAX (OF n CLOSELY ASSOCIATED STALKS)
(G)

FIG. 8
BAMBOO ANALOGY

once again have the solid column and we have already discarded that! It seems now, that we have contradicted ourselves!

To extricate ourselves then, we have a 'play' left and that is, as we pull the diaphragms closer together we must *watch the skin or tube*. There must be a nice balance between diameter of stalk, thickness, spacing or rings and length of stalk and not the least—the load.

The bamboo then points the way to proper making of a column, beam, bulkhead or deck.

Bulkheads, Decks and Shell.

As we already know the ship is a girder supported to a varying degree on a medium. When the ship hogs, the upper longitudinal fibres are columns in tension and the lower longitudinal fibres are columns in compression, and vica versa when the ship sags. Obviously also, the longitudinal fibres at the same time are loaded as beams.

Bulkheads are preponderantly adjacent columns, and columns only until subjected to liquid or bulk pressure; then they become adjacent beams acting in unison, transferring stresses to restraining decks. The decks in that case behaving as columns once more, either in tension or compression. Similarly, and with some modification the decks may be considered to behave.

The shell and continuous longitudinal bulkheads are in themselves girders for longitudinal integrity, and they too act as columns and beams.

The various main structural elements of a ship then, have multi-functional value or should have! It so happens that structural failure on ship board is due to an element—a beam, a girder, bulkhead or deck having possibly all the multi-functional requirements but one, and it fails in that one, and sometimes in more than one.

If we can open up the stress paths, and fair in some reasonable curves for these paths, we can make most of the parts more nearly multi-functional without undue hazard.

The Present Ship's Structure.

Primarily, because of economy in construction costs, ship construction is subjected to angular connections and to the use of standard mill sections. Both construction method, and the very mill sections themselves make it extremely difficult to avoid stress concentration.

The present expedient for changing direction of stress is the heel or beam bracket. That expedient plus the usual use of standard sections still permits a preponderance of mass to one side of the neutral axis of a design increment.

In Figure 9. A bracket modification is illustrated with the tendency shown to 'bring down' the neutral axis.

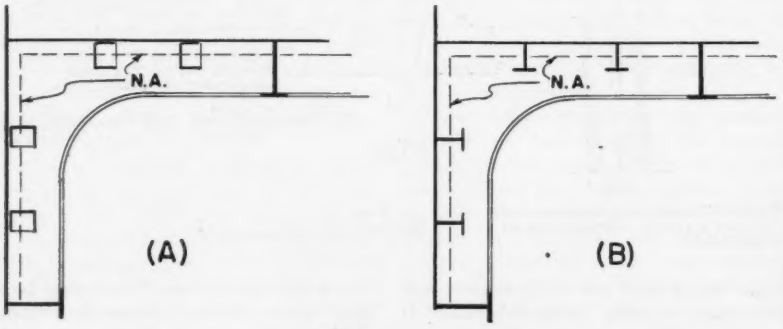


FIG. 9

GRADUAL LOWERING OF NEUTRAL AXIS SHELL, BULKHEADS AND DECKS— MERCHANT HULLS

On Naval vessels and heavier than air craft we have gradually increased the use of flat bar as face bars of stiffeners, sometimes using standard Tees or split beam sections. The idea has been carried out for beams, girders, transverses as well as stiffeners. One notes the gradual increase of the total area of stiffening faces, and indeed papers have been written about the perecentage of face area relative to the panel or plate area. As that 'percent' increases, the neutral axis of any increment approaches a point midway between the outer fibres as in Figure 10.

It must dawn on us that we are actually groping toward double panel or cellular construction. We do, of course, resort to double panel construction on shipboard, such as certain types of metal hatch covers, but they are not a contribution to the ship's girder.

There are now mill facilities available, so adaptable to modern production of weldments that the production of large double panel increments are economically practical. The 'strip mill' together with the better understanding of electric welding at higher speeds and at temperatures above room temperature will lead to the actual production of such panels.

The first step would or should occur immediately after the last stand in a strip mill for 'heavy' requirements, and in the tube mill for intermediate requirements. The light materials would have their start in the strip mill on light gauge operations.

In all types the operation would be on a continuous basis and fully automatic with the usual provision of 'flying cutters'. The panels should be as large as present transport methods and facilities can manage.

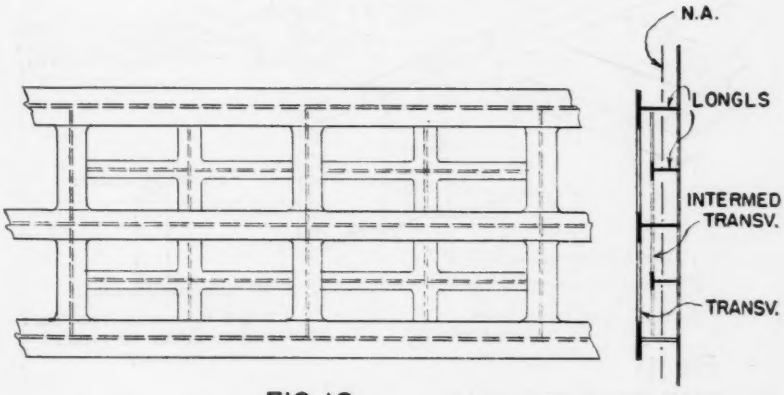


FIG. 10

APPROACHING DOUBLE PANEL CONSTRUCTION BY EXTENSIVE USE OF FACE BARS—INCREASINGLY MORE EXPENSIVE FIELD AND SHOP ASSEMBLY—TENDENCY ON DESTROYERS SUBMARINES AND HEAVIER THAN AIR CRAFT

THE SOLUTION (THE UNIVERSAL STRUCTURAL UNIT).

We have narrowed down to cellular construction with the neutral axis of all increments desirably midway between the outer fibres of increments.

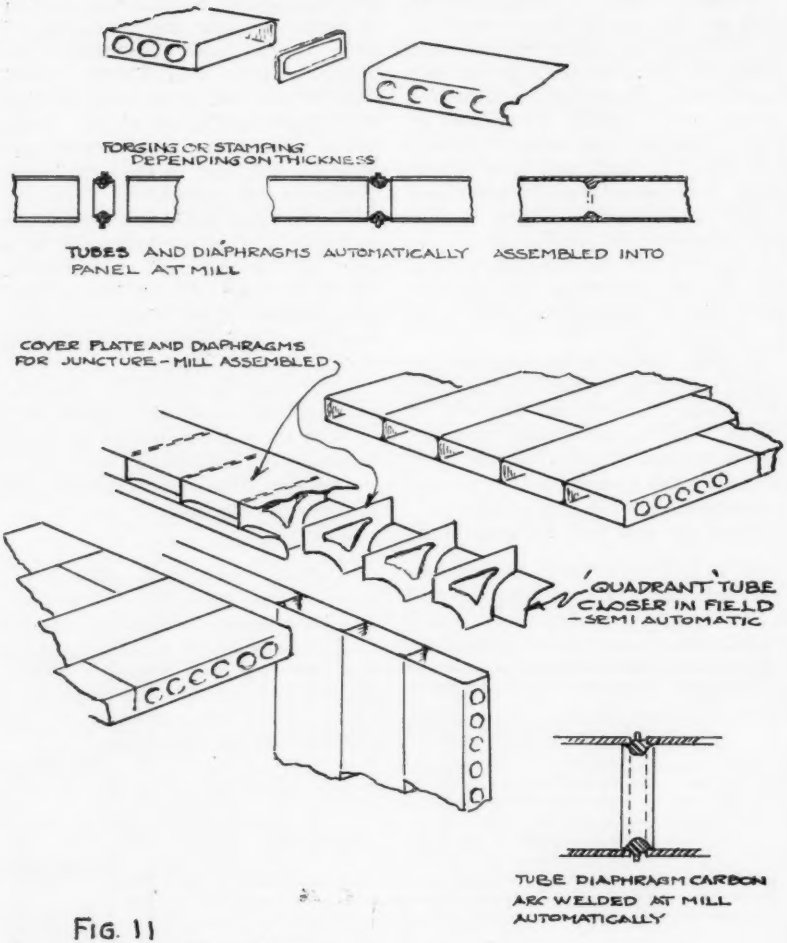


FIG. 11
BULKHEAD AND DECK SCHEME

That the increment be universally applicable is important. It remains to properly balance the paths of stresses, section depth, section width, thickness and spacing of transverse diaphragms.

In doing those things we can meet the requirements set forth by the mathematical prediction and the directives of the welding engineer with a minimum of mechanical and metallurgical notching effect.

Figure 11. Portrays rectangular tubes with rounded corners for deck and bulkhead scheme with the tubes having had the diaphragms introduced assembled into panels on a continuous basis.

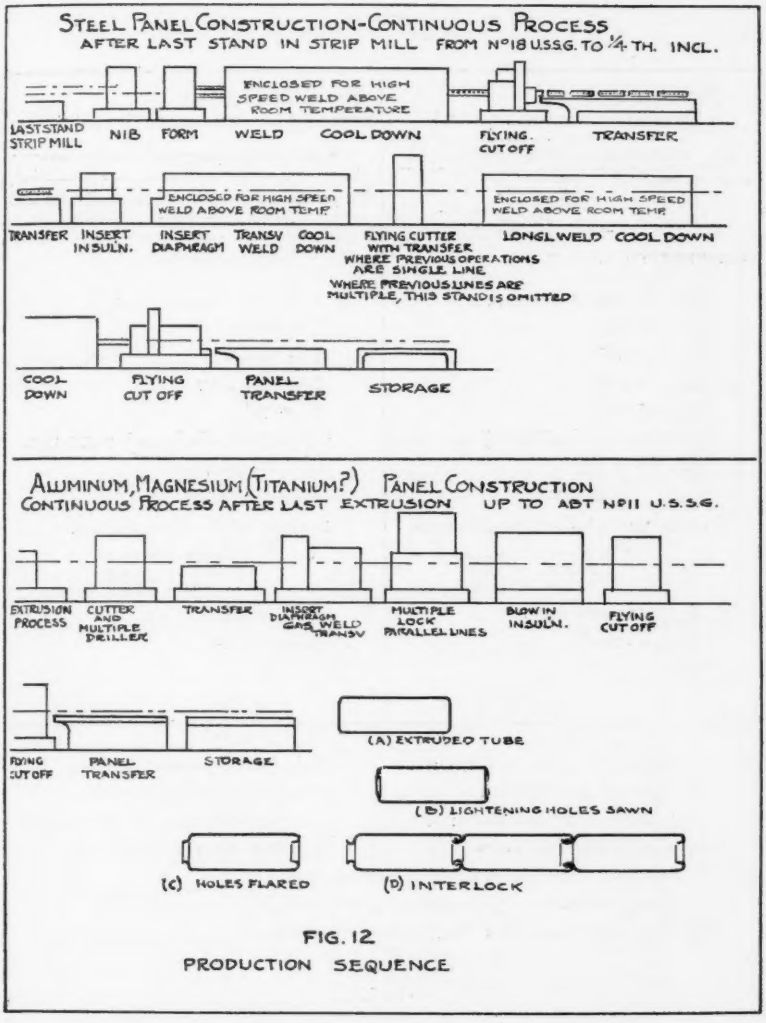


FIG. 12.
PRODUCTION SEQUENCE

Figure 12. Shows the method of production in outline.

Figure 13. Is a comparison of deck increments of a destroyer.

Figure 14. Shows a room arrangement.

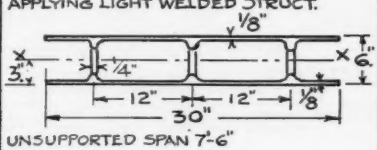
Figure 15. Shows some characteristic details and a typical heat transmission condition.

DECK INCREMENT
2100 TON DESTROYER (U.S.N.)

UNSUPPORTED SPAN 7'-6"

A = 13.746 SQ. INCHES
I = 42.44
r_{x-x} = 1.755
S_{x-x} = 7.895
Y = 5.375"
WT PER LIN. FT. (30" WIDE) 46.75 LBS.
$\frac{l}{r} = \frac{90}{1.755} = 51$
PANEL 7'-6" x 2'-6"
LOAD W: 4'-0" HT. GREEN SEA @ 512" P.S.F. = 9600
STEEL = 351
W = 9951 LBS.
$f_{MAX.}$ = 14200 P.S.I.
D_{MAX} = .074"

SAME DESTROYER
APPLYING LIGHT WELDED STRUCT.



UNSUPPORTED SPAN 7'-6"

A = 11.25 SQ. INCHES
I = 74.88
r = 6.45
S_{x-x} = 24.967
Y = 3"
WT. PER LIN. FT. (30" WIDE) 39.75 LBS.
$\frac{l}{r} = \frac{90}{6.45} = 14$
PANEL 7'-6" x 2'-6"
LOAD W: 4'-0" HT. GREEN SEA @ 512" P.S.F. = 9600
STEEL = 298
W = 9898 LBS.
$f_{MAX.}$ = 4470 P.S.I.
$D_{MAX.}$ = .042"

A AREA OF TOTAL SECTION
I MOMENT OF INERTIA
r_{x-x} RADIUS OF GYRATION
S_{x-x} SECTION MODULUS
Y DISTANCE FROM NEUTRAL AXIS TO EXTREME FIBRE
W TOTAL PANEL LOAD, UNIFORMLY DISTRIBUTED
$\frac{WL}{8}$ ASSUMED FOR M_{MAX} FOR READY COMPARISON

D_{MAX} MAXIMUM DEFLECTION
f MAXIMUM FIBRE STRESS
l LENGTH OF COLUMN (SPAN) IN
$\frac{l}{r}$ RATIO OF SLENDERNESS

FIG. 13
COMPARITIVE CHARACTERISTICS 2100 TON DESTROYER

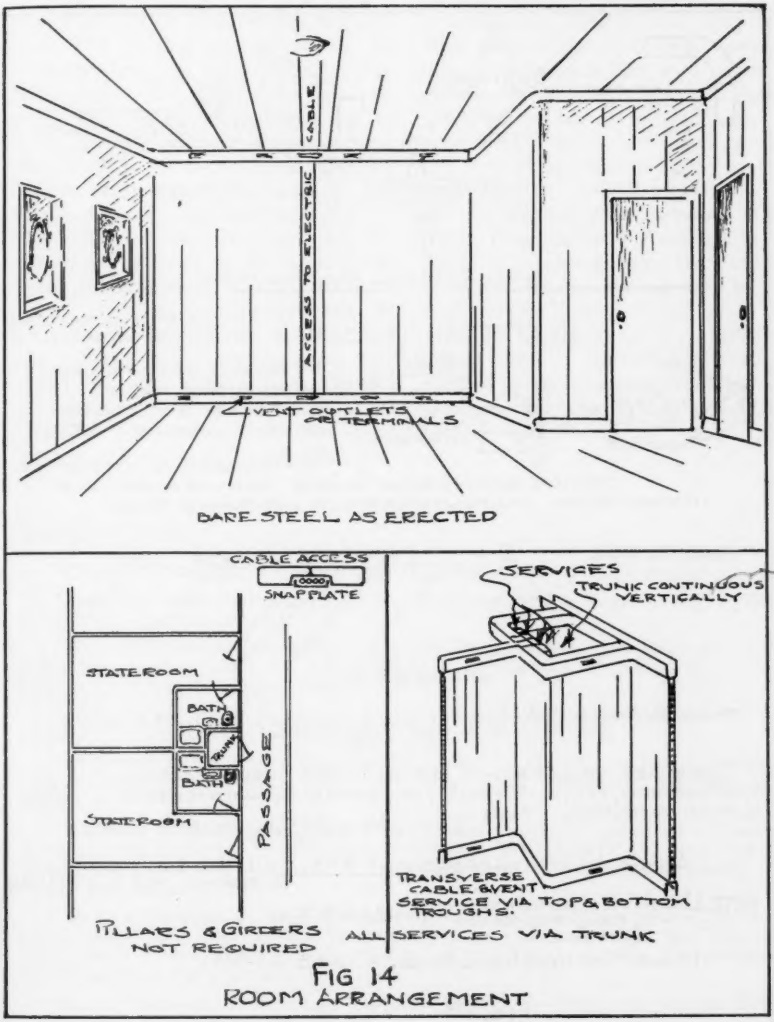
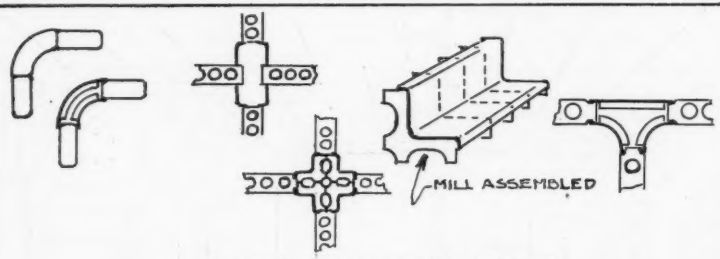


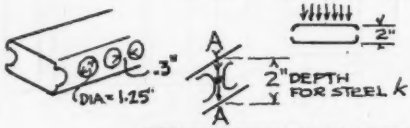
FIG 14
ROOM ARRANGEMENT



SOME TYPICAL CONNECTIONS

HEAT TRANSMISSION

USING 9"x 2" O.DIM. TUBE BY 1/16" TH. WALL



U = OVERALL RATE OF TRANS-MISSION, FACE TO FACE PER DEGREE Td PER SQ. FT. OF PANEL PER HR ~B.T.Us

Td = TEMPERATURE DIFFERENCE.

TRANSMISSION IS FACE TO FACE PLUS TRANSMISSION THROUGH STEEL ELEMENTS BETWEEN LIGHTENING HOLES

FACE TO FACE $U = \frac{1}{.166 + \frac{.0625}{310} + \frac{2}{.24} + \frac{.0625}{310} + .606}$

(.166: O.S. AIR FILM; .0625/310: OUTER STEEL; 2/.24: ROCKWOOL 4# PER CU. FT.; .0625/310: INNER STEEL; .606: I.S. AIR FILM)

$= .1098$ B.T.Us

THROUGH STEEL A-A $U = \frac{1}{.166 + \frac{2}{310} + .606} = 1.294$ B.T.Us

THERE ARE 20.645 POINTS PER SQ. FT. FOR THROUGH STEEL TRANSMISSION, (EACH .3"x.0625") OR .04625 SQ. INCHES PER POINT = .9548 SQ. INCHES; THEN $U = 1.294 \times \frac{.9548}{144} = .00858$ B.T.Us

ADDING; $U_2 = .1098 + .00858 = .11838$ B.T.Us PER SQ. FT PER DEG. F. Td PER HR FOR 9"x2"x1/16" TUBE

NOTE U FOR 1/4" STEEL PLT + 2" ROCK WOOL + 1/16 STEEL SHEATHING + 2 1/2" ANGLE } = .1366 B.T.Us

U FOR 1" STD. HARD INSUL. PANEL = .1345 B.T.Us

FIG. 15

Corrosion.

It will be said that corrosion will have a very undesirable influence on completely enclosed structure. Corrosion should be no insurmountable problem. The general laws governing the different kinds of corrosion are fairly well understoood. There are the various steels having resistance to corrosion, plastic coatings, metallic coatings, the system of bonderizing and the system of carronizing.

The use of a low melting temperature alloy of Bismuth, Zinc and Tin which will be used in the assembly of tube panel elements will furnish protection for the life of a vessel. It should be noted that the voids within the panels will contain insulating material thereby reducing considerably the amount of imprisoned air. The use of non hygroscopic insulation will further reduce the chances for corrosion.

Metal hatch covers on some British cargo vessels have shown very good resistance to erosion and corrosion. Most of them are of cellular construction, plug welded and have approximately 14 U.S.S.G. skin thickness. Many are still in use after 20 years in service as 'Tween deck hatch covers.

The packing box industry has long since adopted cellular construction in paste board, and some forms of 'honey comb' construction are applied in the plastics. It seems then, that cellular construction in steel is not beyond practical application. Certainly it will give the engineer an opportunity to design ships, houses and railway coaches by direct calculation and leave empirical formulae in the limbo where they should rest.

BIBLIOGRAPHY.

"Membrane Analogy"—"Strength of Materials"—Part II—Timoshenko page 623.

"Reduction of Deformation of Welded Structures by One Side Welding"—"Metals and Alloys"—January 1940.

"Experiments with Single Sided Welding of a Vessel's Hull Plating."—V. K. Poleshink—Soudostroienie No. 2 1939 PP 94-98—In Russian, Abstracted in "Welding Lit. Review." Vol. 1—August 1939, P. 112.

"Strength of Ship's Structures Under Compressive Stresses"—E. H. Mitchell, Member, in "Transactions of Institute of Naval Architects" 1936.

"Fatigue Problems in Structural Design"—Article in "Metals and Alloys" December 1939—Concluded from Nov. 1939 Issue,—By A. V. Karpov—Consulting Engr. Pittsb. Pa.

"Buckling of Rectangular Plates: The Critical Value of Compressive Stress"—Solution by G. H. Bryan—London Math. Soc.—Proc. Vol XXIV, P. 54, -1891—See also Timoshenko's Table of Contants for Calculating Compressive Stress in Simply Supported Rectangular Plates.

"Relative Strength of Conventional and Stressed Skin Structures"—"Mechanical Engineering," Nov. 1940,—Article by Messrs, Beemer—Lindvall, Stoner and Van Dorn —Excerpted from Original Article "A Fundamental Development in Suspension and Construction of Railway Cars."

Euler's Theory-Long Columns—"Applied Mechanics"—Fuller and Johnson Vol. II —"Strength of Materials" -1919 P. 348.

"Stress Concentration"—Timoshenko—"Strength of Materials" Part 2—Advanced Theory P. 615.

"Theory of Uniform or Simple Bending"—"Applied Mechanics" Vol. II of—Strength of Materials 1919 P. 107.

"Resistance Welding"—"Materials and Methods Manual" Nov. 1947.

"'Metallizing"—"Materials and Methods"—Sept. 1948.

"Report on the Fracture of Metals"—Part I—Contract Nobs 24041—Between—Carnegie Inst. of Technology and the Bureau of Ships.

"Report on the Fracture of Metals"—Part II—A Supplementary Survey—Under Contract No. Nobs 24041, Under Auspices of the Office of Naval Research Conducted under Contract No. NG Or 1-155 with Pennsylvania State College.

"Survey of Automatic Arc and Gas Welding Processes as used in the Automotive Industry"—"The Welding Journal"—January 1948—Survey Conducted by the American Welding Society by its Members.

"Gage of the Future"—"Steel Ways"—Publication May 1948 by the "American Iron and Steel Institute"—Article by Mr. Gold V. Sanders.

"Le Tourneau Cellular Shop"—"Engineering News Record" Oct. 10, 1940 Page 48.

"Technical Progress Report of the Ship Structure Committee"—to the Secretary of the Treasury.—Welding Research Supplement of the 'Welding Journal' of the A.W.S.

PRACTICAL SHIP DEHUMIDIFICATION—The Effect of Water Vapor Diffusion and Adsorption.

BY ALBERT S. GATES, JR.

THE AUTHOR.

ALBERT S. GATES, JR. came to the Bureau of Ships in 1940. He has been responsible for dehumidification since 1946 and also handles submarines air conditioning and purification. He served two years each at the Naval Shipyards Portsmouth, N. H., and Boston, and six years with the Boston and Maine Railroad. He obtained his engineering education at the University of Maine.

At the height of its power during the last war, the United States Fleet was greater than the Navies of the World, enemy and friendly combined, and was the greatest Naval force ever assembled. It was several times larger than the greatest force which could possibly be kept in service under any conceivable peacetime situation. The Navy's long range planners realized that this time, when hostilities should cease, it would not be sufficient to scrap a few of the older vessels, drastically reduce the crews, and drastically curtail Fleet operations, but that demobilization of a great number of the vessels themselves would be necessary. Their problem was to accomplish this demobilization with no unnecessary dissipation of the military potential and the investment, represented by the vessels affected.

By 1944, it had been decided to dehumidify the ships. It was known that dehumidification would retard the deterioration of the interior of the vessels and would help in the preservation of machinery and equipment. It was not then realized just how effective, simple and inexpensive dehumidification would finally prove to be. More important, however, several groups of questions remained to be answered before dehumidification of inactive ships would become a technique which could be economically applied on a large scale by widely separated Naval activities to a great number of different kinds of vessels. The experiments described below were conducted to answer, by a practical demonstration, some of the most important of these questions: those concerning water vapor diffusion and adsorption. We shall first describe the experiments, then discuss the questions and the answers which were obtained.

A small vessel which had been constructed as a specialized auxiliary proved on her completion not to be required for her original duty, and was assigned as a floating laboratory of preservation methods and equipment. It was already under dehumidification using the pilot models of dehumidifiers which were later developed as standard Navy equipment. The compartments in which the diffusion experiments were

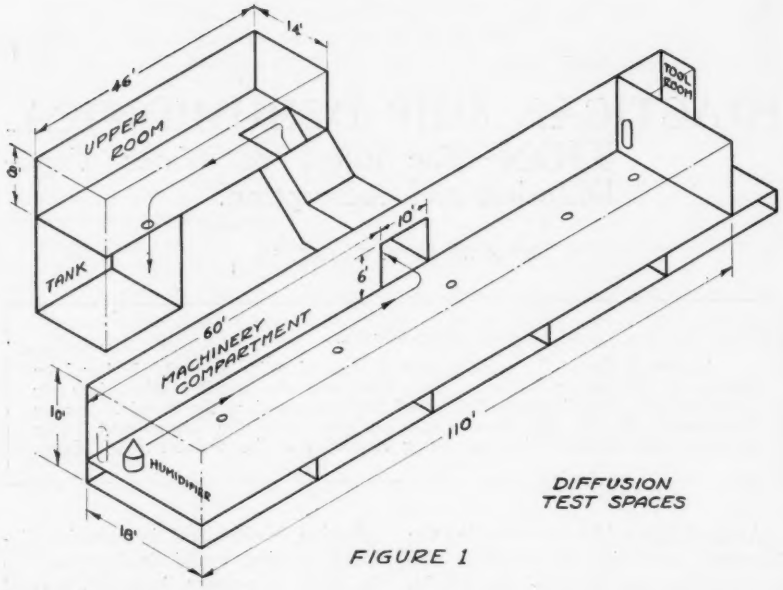


FIGURE 1

made, their principal dimensions, and the accesses are shown in Figure 1. The three nearer boundaries of the rooms are shown in phantom view and some liberties have been taken with the scale, in the interest of greater clarity. The total volume involved is about 32000 cubic feet.

Water vapor was added to the compartments when necessary during the experiment by means of a humidifier located near the after bulkhead of the machinery compartment. This device, shown in Figure 2, consisted of an inverted cone wick of rough toweling on which water could be dripped, suspended over a 2 KW electric heater. Dry bulb temperatures were taken from the temperature records of a number of hythergraphs systematically distributed throughout the compartments, on a few of which glass-alcohol thermometers were also attached for a check on accuracy. One can be seen in Figure 2.

Humidity measurements were taken by reading dew points with the ingenious device shown in Figure 3. This instrument was christened the "Humidity Analyser" by the experimenters. It comprises a slot, called the "view box" in Figure 3, through which the air sample is drawn from right to left. The front of the slot is glass and the back is a thin plate of highly polished metal, the back of which is bathed by a stream of liquid which flows from left to right. The temperature of the plate at any point is indicated by the 24 thermometers shown. A fairly uniform temperature gradient of about 15°F, which brackets the dew point of the passing sample of air, is established in the liquid by means of refrigeration and electric heat. The exact dew point of the sample is read on the thermometer which coincides with the boundary of the resulting fogged area on the mirror surface of the metal plate. The apparatus is purged with

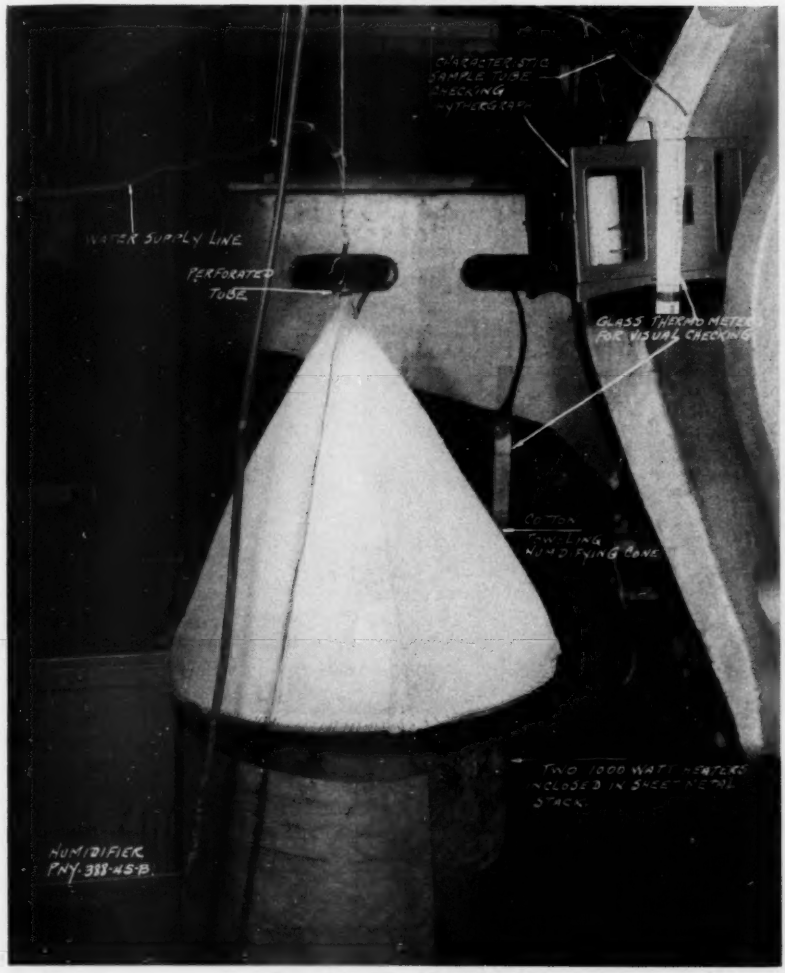


FIGURE 2.—The Humidifier.

dry air after each reading. An elaborate system involving over 8000 feet of copper sampling tubes, a "switchboard" for selecting the proper sample, and an air pump which could deliver the sample, in 1½ seconds, were installed. Figures 4 and 5 are views of these arrangements. During the tests the dew points of air samples drawn from 105 locations in the test compartments were periodically measured with this instrument, at an average rate of less than one minute per sample.

The experiment was started at 6:00 A.M. on December 19, 1944 and required a period of 96 hours divided into four phases: a blank run of 24 hours, a humidifying run of 18 hours, a stabilizing run of 18 hours and a dehumidifying run of 36 hours. At the

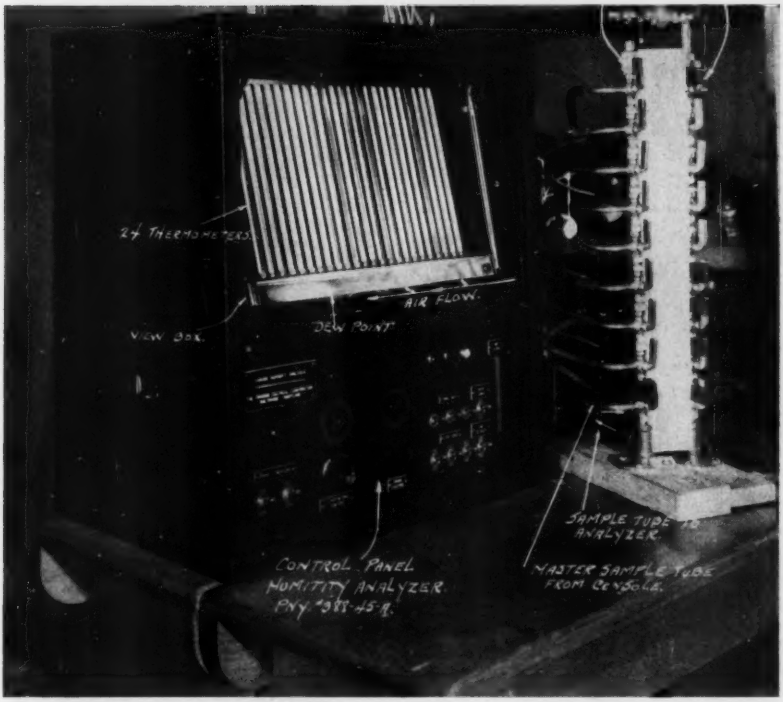


FIGURE 3.—The Humidity Analyser.

beginning of the blank run the compartments were vacated and all openings to the weather and the remainder of the vessel and all connections to the dehumidifiers were sealed, but the 16″ x 29″ manholes to each of the tanks remained open. At the end of the blank run, the entire space had equalized at about 30 percent humidity. The humidifier was then operated to deliver about 0.8 lbs. of water vapor per hour. After two hours, the rate was increased to about 1.4 lbs. per hour. This rate was maintained until the end of the humidifying run, when the humidity has risen to 50 percent and about 21 lbs. of water had been evaporated. The humidifier was then secured and the conditions in the compartments allowed to equalize during the stabilizing run. The 26″x66″ door in the after bulkhead of the catapult machinery space was then opened. During the dehumidifying run, the water vapor previously added to the test compartments diffused through this door to the passage aft, which was under dehumidification at about 30 percent humidity by the ships dehumidifying machines. At the end of the 36 hours of the dehumidification run the test was secured. No personnel were allowed in the test compartments during the entire test.

The diffusion questions were directed toward defining the boundary between inadequacy and overdesign in three features of the ship dehumidification system. First, would multiple dry air outlets and facilities for air circulation be necessary in each compartment.

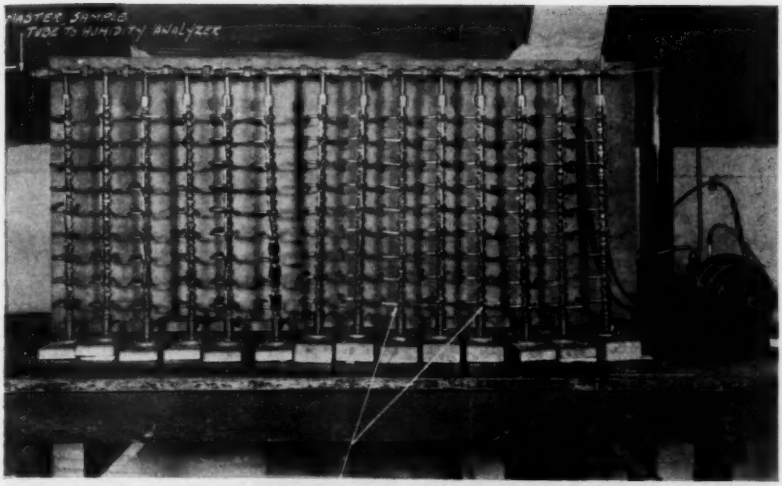


FIGURE 4.—The Sampling Tube Switchboard.

or would diffusion be sufficiently rapid to take care of distribution in a compartment? If so, how large a compartment? Second, must a dry air input be provided for every compartment, or would diffusion from dead end spaces through doors, hatches and manholes to compartments having a dry air supply permit the use of a simpler and less costly dry air distribution system? Third, there would be diffusion of water vapor from the weather through the inevitable cracks and apertures. What magnitude of load would such diffusion place on the dehumidifying equipment? What measures to improve the weather boundary tightness would be feasible and economically justified? In the light of certain previous experiments on CO_2 produced by men in sealed compartments, during which it was noted that fairly high concentrations would diffuse almost instantly to a low value when the seal was broken, it was expected that diffusion rates would be high.

Efforts had of course been made, before the experiments, to answer these questions from information already available. The various authorities are in substantial agreement on the rate of water vapor diffusion in air. The coefficient of diffusion is given by Handbook of Chemistry as 0.239 sq. cm/sec, and by the International Critical Tables as 0.220 sq. cm/sec. These values apply in the equation

$$\frac{dq}{dt} = -K \frac{dc}{dx} \, dydz$$

where q is the mass of water vapor diffusing betwen two points
t is time
K is the diffusion coefficient
c is the difference in volume concentration of vapor at the two points
x is the distance between the points
y, z are dimensions of the diffusion path normal to the flow.

The equation can be used to predict the amount of water vapor which will diffuse by steady flow along a defined path between two points at different humidities. It will also give the time to reach a uniform humidity in a vessel of simple shape with a known humidity

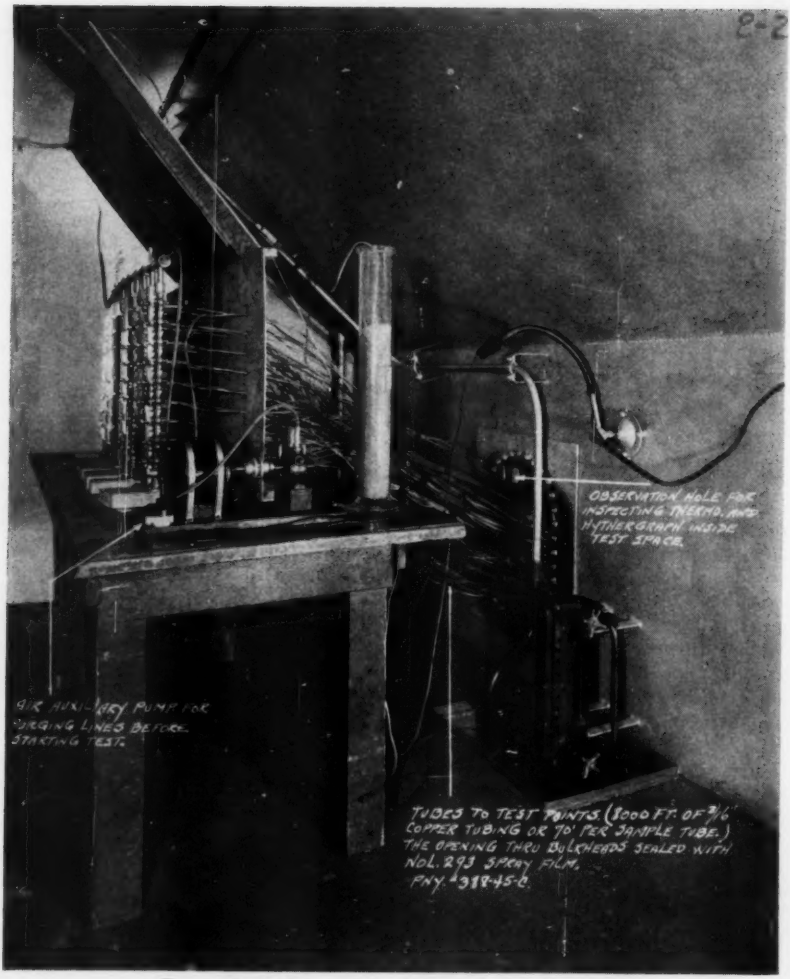

E-2
OBSERVATION HOLE FOR INSPECTING THERMO. AND HYTHERGRAPH INSIDE TEST SPACE.
AIR AUXILIARY PUMP FOR PURGING LINES BEFORE STARTING TEST.
TUBES TO TEST POINTS. (1000 FT. OF 3/16" COPPER TUBING OR 70' PER SAMPLE TUBE.) THE OPENING THRU BULKHEADS SEALED WITH NOL. 293 SPRAY FILM. FNY. 1318-45-B.

FIGURE 5.—A View of the Sampling Arrangements.

gradient. But if it is applied to even the simplest shipboard situation it leads only to new questions: How can the two points be chosen so that the results will be significant to the practical problem? What is effective area of the diffusion path, particularly where it passes through a door or hatch? How can the values of q and t be interpreted in terms of the rate at which water vapor must be removed from a ship? The practical interpretation of any possible calculation was so dubious, and the cost of any misconception applied to the whole Reserve Fleet could be so high, that the expense of the full scale tests was felt to be fully justified.

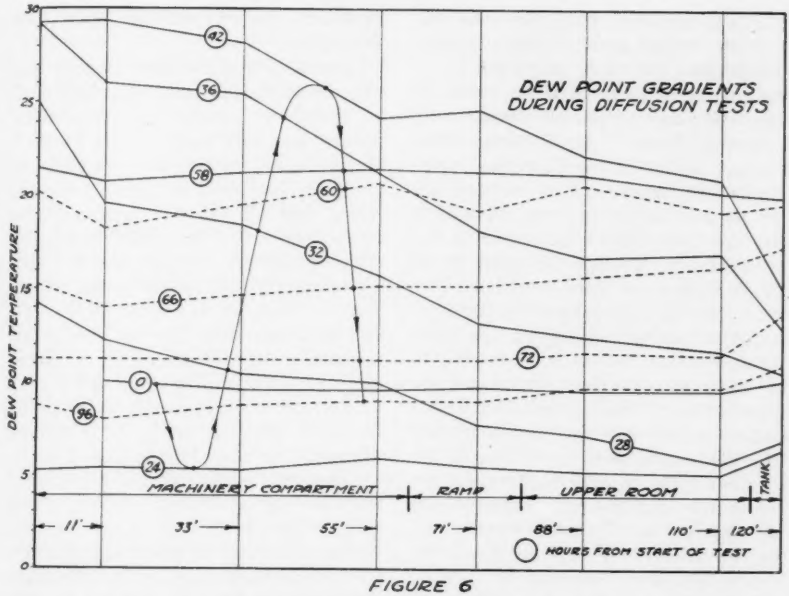


FIGURE 6

Figure 6 is an attempt to organize, in such a way as to bring out the diffusion effects, a very small portion of the data taken. Each curve shows for the particular time circled the dew point at various locations along the particular diffusion path shown on Figure 1: from the humidifier through the machinery compartment and the ramp to the upper room and thence through the 16″ x 26″ manhole to the tank below, a distance of about 120 ft. Each dew point shown is the average of the readings of all the samples in a plane across the compartment normal to the flow, so that it represents the mean dew point across the compartment. The lines for the diffusion run are shown dotted for greater clarity. The S shaped arrowed line is an attempt to show more clearly the time relation of the various curves.

During the humidifying run from 24 to 42 hours, the dew point rose from about 5° to an average of about 25°, and the dew point gradient from the humidifier to the end of the upper room was approximately 15°. At 58 hours, 16 hours in the stabilizing run, this gradient had practically disappeared. When the diffusion run was started at 60 hours a dew point gradient was established in the opposite direction, but never exceeded about 4°. At 72 hours the gradient had practically disappeared, and at 96 hours, the end of test, the compartments had come to equilibrium at about 8°, the dew point then prevailing in the passage. The tank, which followed the changes in the adjacent compartment but lagged them as much as 8°, was typical of all the tanks in the test space. This data was enough to show that, with humidity differences within a range acceptable for preservation, the rate of water vapor diffusion would be appreciable. Because of the surface adsorption effect to be discussed later, the data cannot be used to check our diffusion formula and coefficient.

It was apparent from the tests that a single dry air supply, without special provisions for air distribution or special attention to location, would be ample for any shipboard compartment however large. Any compartment through which it should prove inconvenient or impractical to arrange for dry air circulation, could be allowed to dry by diffusion to an adjacent dry compartment through an opening no larger than an ordinary manhole. The type plans of the dehumidification installations were prepared on this basis. In most ships, the firemain proved to be the most convenient system for the distribution of the dry air, since it has outlets in all but a few compartments. The air was allowed to return to the dehumidifying machine through accesses and passageways. Compartments not having a firemain outlet, but located adjacent to the path of the return air, were allowed to dry by diffusion. To provide for a few remaining compartments, the firemain was extended as a dry air distribution system by means of collapsible canvas air hose.

The behavior of the dew point in the tool room gave the answer to the third principal question, concerning the rate of diffusion of water vapor to be expected from the weather through small cracks and openings. The watertight door on this space was clamped tight shut with five of its six clamp dogs, but the sixth dog and its spindle were removed, leaving a hole one inch in diameter between the tool room and the machinery compartment. If there had been any significant diffusion through this hole during the test, the tool room dew point, like that in the tanks, would have followed that at a nearby point in the adjacent compartment, except that the lag would have been greater in the tool room because of the smaller hole. There was no such effect. This test indicated diffusion from the weather would be no serious problem, and experience has confirmed it.

Considerable pains are taken to seal the vessel. On all large openings such as those for ventilation, sheet metal covers are tack welded and caulked. Drains and vents are capped or plugged. Doors and hatches are dogged down, and all but one necessary for access are sealed by caulking, tape or sprayed plastics. The tightness is tested by operating one of the ships supply fans in such a way that all the air delivered must leave through the leaks, measuring the airflow and the resulting static pressure, and computing the area of an orifice which would develop the observed pressure with the observed airflow. The ship is considered to be satisfactorily sealed when the computed area of the equivalent orifice is not greater than 1/10 of a square foot per 100000 cubic feet of dehumidified space.

The load remaining for the dehumidifier when the ship has once been dried out consists of the "breathing" load caused by air temperature changes; the infiltration load caused by wind pressure, and the diffusion load. It is impractical to measure the relative size of these loads, but by inference from the observed behavior under varying conditions, the diffusion load appears to be an insignificant part of the total. For example, in rainy weather, when the load might be expected to rise, because of the absence of the sun the breathing load is reduced and the load on the dehumidifier actually falls. From the effect of varying wind conditions on the total load, deductions can similarly be made about the magnitude of the infiltration load. From such indications these two loads appear to be almost the total load. The effect of using the accesses also gives an indication of diffusion. If the ship is entered excessively, as for example on the ship used for instruction in the preservation procedures, the additional load is reflected in higher humidities and opera-

tion of the dehumidifier for a greater portion of the time. With the ship entered circumspectly and not more than 2 or 3 times daily, however, the additional load due to diffusion is too small to be detected. The significance of diffusion while the access is open was most aptly expressed by one instructor, who explained that the dehumidified ship is analagous in this respect to a refrigerated space, where one enters and closes the door as often as necessary, but avoids unnecessary entry and never leaves the door standing open.

For lack of a better term we have used the word "diffusion" rather loosely to describe phenomena which no doubt resulted partly from air convection. Since the real interest was the movement of the water vapor, not the precise cause of the movement, the concern with convection was to avoid any convection effect during the tests which would not also occur in practice on the inactive vessels. The tests were sufficiently representative in this respect of the actual conditions later encountered, except that they were conducted in winter. Partly for this reason the experimenters qualified their conclusions by the recommendation that similar tests be run during the summer, but summer tests were never run. It has been found in practice that, on ships whose air distribution is balanced in winter to maintain fairly uniform humidity throughout the ship, rebalancing becomes necessary for summer conditions. Once a satisfactory summer balance is achieved, however, it will hold for summer and winter. The winter balances, which were relatively easy to achieve, apparently depend to a considerable extent on the mixing effect of the convection currents induced by the flow of heat upward through the vessel from the sea. The downward heat flow of summer inhibits the convection, so the summer balances are dependent primarily on diffusion and are little affected by further changes in the convection rate.

No questions on surface adsorption of water vapor had been listed for answering during the tests, because no appreciable effect on dehumidification was anticipated. Absorption, which concerns water in the liquid phase, had of course been eliminated, but surface adsorption is a phenomenon of the vapor phase and is always present. It may be said to result from the balancing between the kinetic energy of the water molecules in the surrounding vapor and the forces of attraction of water molecules for each other and for the molecules of the adsorbing surface. A water molecule striking the adsorbing surface without rebounding imparts its kinetic energy to other molecules in the complex of surface molecules and already adsorbed water molecules. Any water molecule which acquires enough of this energy to overcome the attractive forces, escapes from the complex. At equilibrium, the number on unit area which escape or rebound statistically equals the number which strike, and the number attracted remains statistically constant. If now the concentration of molecules in the surrounding vapor, that is to say the dew point, increases, the number of impacts per unit area increases, causing a corresponding increase in the number held in attraction and the number escaping, until all factors again balance at higher values. Thus the amount adsorbed at a given temperature increases with increasing dew point. A change in temperature in the range of weather temperature has little effect on the forces of attraction between molecules, but it directly affects their kinetic energies. The energy necessary for escape is practically constant, but with a rise in temperature the energy of both the adsorbed and the free water molecules increases, so that fewer can be retained on unit area by the forces of attraction, and water vapor is desorbed

until a new equilibrium is reached. Thus the amount adsorbed at a given dew point decreases with increasing temperature.

The combined effect of these relations has been tested in metal packages sealed with various contents and exposed to the weather. In an empty package, the ratio of adsorbing surface to the volume is small, so that the vapor concentration in space is fairly constant in spite of the changes with temperature of the amount adsorbed on the surfaces. The dew point is nearly constant but the humidity varies widely and in the direction opposite to the temperature. If the package contains considerable adsorbent material such as cardboard cartons and wood, or an amount of desiccant, the ratio of adsorbent surface to volume is very high. A rise in temperature causes desorption and a consequent increase of vapor concentration in space, and conversely. In such packages the dew point varies directly with the temperature, but the per cent humidity remains nearly constant. With packages between these extremes, both dew point and per cent humidity vary with temperature: the greater the amount of adsorbent capacity, the greater the variation in dew point and the less the change in per cent humidity.

The capacity of practically every substance to adsorb water vapor on its surface to some degree has long been well known. Some substances of a porous nature which thus have a relatively high exposed surface area, and some between whose molecules and those of water vapor there are strong attractive forces, exhibit this property to a high degree and are called hygroscopic. The desiccants head this list, but paper, leather, glass wool, textiles, hemp and many other substances are included. Wood, for example, can adsorb about one forth as much water vapor as an equal weight of silica gel and nearly as much as activated alumina.

There was very little material of a hygroscopic nature in the test compartments. Since the ship had never seen service and the compartments had been under dehumidification the surfaces were new, clean and dry, and would have the property of adsorbing water to a very small extent only. The amount of adsorbed water on one square foot of a bulkhead in equilibrium at 30% humidity would be very small indeed. In coming to equilibrium with 50% humidity it would adsorb an even smaller additional amount. In a space the size and nature of a ship compartment, however, the total area is sufficiently great to make surface adsorption an important factor. In the machinery compartment alone the area of the plating of the boundary is about 6500 square feet and that of other structural members may be half as much, not to mention the surfaces of machinery, pipes, cables and similar items. Any roughness of these surfaces increases their effective area. It would be impossible to estimate the total effective area of surface in the test compartments, but it was surely very high.

When the effects of adsorption on these surfaces began to become apparent, which they did early in the tests, they were both surprising and puzzling. First, during the first 24 hours, with the test compartments sealed and no water vapor being added or subtracted, the dew point for no apparent reason dropped in all compartments from about 10° to about 5½°. This change was shown to follow a fall in temperature during the same period. Second, during the humidifying run, it was found necessary to operate the humidifier for 18 hours and to add 21 pounds of water vapor to the space in order to raise the humidity from 30% to 50%, although it had been planned to accomplish the same result in about

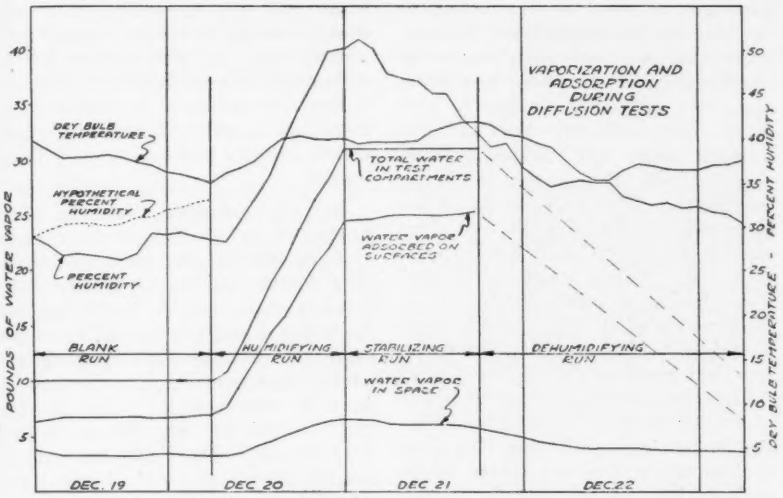


FIGURE 7

5 hours with only 4 pounds of water vapor. These tests were the first conducted both on such a scale that the total adsorption on all the surfaces was of appreciable magnitude, and in such a manner that the adsorption effects could be distinguished from other humidity phenomena.

The curves on Figure 7 give an indication of the relative magnitudes and rates of the vaporization and adsorption effects. The curve "Water Vapor in Space" is computed from the mean of the observed dew points and may be considered the result of direct observation. The curve "Total Water in Test Compartments", is valid only to show the differences in total water resulting from the humidification, since it is based on the assumption that the sum of the water adsorbed on the surfaces in the test compartments plus the water vapor in the space of the compartments was ten pounds at the start of the test. This assumption is wholly arbitrary; it is obvious that many times this amount was already adsorbed at the start of test. The curve "Water Vapor Adsorbed on Surfaces" is plotted by subtracting the values of the curve "Water Vapor in Space" from the values of the curve "Total Water in Test Compartments", and is also valid only for the differences it shows. From the time of opening the test spaces at the start of the dehumidifying run the latter two curves are shown dotted, since there is no longer any valid means of showing even their relative locations. The "Dry Bulb Temperature" and "Percent Humidity" curves show the mean of the observed values throughout the compartments.

The dotted "Hypothetical Percent Humidity" Curve is based on the assumption of constant dew point. The difference from the actual percent humidity curve gives an indication of the increased adsorption during the blank run due to the fall in temperature. The increase of adsorption with increasing vapor concentration is strikingly shown during the humidifying run. In coming to equilibrium with the rising concentration, the surfaces adsorbed vapor from the space nearly as fast as it was added. The desorbing effect of the slight rise in temperature

cannot be detected on the curves. At the end of the humidifying run, the adsorption was only slightly below equilibrium with the vapor concentration then prevailing, indicating that water vapor could have been added at a much faster rate. Adsorption continued slowly during the stabilizing run and there is no evidence that it had completely leveled off at the end. Concerning the desorption, it will be noted that the "Water Vapor in Space" curve had substantially leveled off by the morning of the 22nd, which might be taken to indicate that desorption was then complete. On the other hand the temperature drop during the middle of the day on the 22nd coincides with a rise in humidity, indicating that little surface adsorption was taking place, perhaps because the adsorbed water content still was higher than for equilibrium with the vapor concentration. The tests give no conclusive evidence of the desorption rate.

Although it had not been considered before the tests, the surface adsorption was taken fully into account in subsequent dehumidification work. It is hardly possible to overestimate the importance of its effect on practical ship dehumidification. In the first place it probably delayed the development of the technique for many years. We now know that an initial drying period of 6 weeks to 3 months, during which the dehumidifier works continuously at high capacity to remove great quantities of adsorbed water from the surfaces, must take place on any ship, and that thereafter the normal load of infiltration and breathing can be removed easily. The pioneers who about 1925 were trying to dehumidify ships statically with lithium chloride, were forced to abandon their experiments, though not their convictions, for lack of time and funds, before having discovered the great difference in magnitude between the initial and normal loads. Had they known this fact they could no doubt have made a more convincing case for dehumidification as economical ship preservation, obtained support for equipment development and continued experimentation and given us the working technique twenty years sooner.

Surface adsorption causes an effect analogous to inertia, which prevents rapid changes in the space conditions. The inertia makes the system independent of peaks in the infiltration and breathing loads. It permits extensive repairs to the dehumidifying equipment providing reasonable care is taken to keep the vessel closed. It makes possible the use of an intermittent control arrangement without adversely affecting the preservation and without waste of power in the dehumidifier.

Adsorption contributes to control simplicity. The dehumidified ship is analogous to a package with fairy high adsorbent capacity, so its percent humidity varies only slightly with temperature. Percent humidity is the humidity quality which can most conveniently be used to generate a control-operating force. Its variations most nearly correspond to the variations in humidity-induced deterioration. These factors make the arrangement of suitable controls a simple and straightforward problem.

The seasonal temperature changes and the surface adsorption phenomena operate together to cause seasonal load variations. In the fall, the decreasing temperature of the surfaces increases their adsorbent capacity, they adsorb much of the water vapor of the breathing and infiltration load, and the dehumidifier operates at its lowest load of the year. In the winter and late spring, the load is stable and low. As soon as the surfaces begin to warm from the rising temperatures of approaching summer, the water vapor

previously adsorbed is released into the space, and, combined with the breathing and infiltration, becomes the dehumidifiers highest load of the year. The summer load is stable, but higher than the winter load because of the greater range of daily temperatures and higher weather dew points.

Acknowledgment.

The full scale dehumidification tests were conceived in the Bureau of Ships. They were carried out by the military and civilian personnel of the Material Laboratory at the Naval Shipyard, Philadelphia, to whom great credit is due not only for excellent performance of the test work but also for many original ideas and suggestions contributed to the test procedure. The Davison Chemical Company, then under contract as consultants, also contributed valuable advice and suggestions. The writer had no duties concerning the dehumidification program at the time of the test and did not participate, but offers this report in response to numerous inquiries.

NOTICE

So little use has been made of the tear sheet which is placed in the back of the JOURNAL to permit readers to suggest titles and authors that it will probably be omitted after this issue.

NATURAL FREQUENCIES—A Method of Solution in Undamped Torsional Vibration Problems.

BY MERL D. CREECH AND JAMES O. MELTON.

THE AUTHORS.

MERL D. CREECH was born in Kansas, 1903; attended the University of Oklahoma 1924-1930; received the degree of B.S. in Mechanical Engineering 1929 and M.S. in Engineering 1930; held Research Fellowship in Department of Mechanics, University of Oklahoma 1929-1930; instructor University of Tulsa, 1931; Development Engineer Black Sivalls & Bryson, Oklahoma City, Oklahoma, 1936-1940; Plant Engineer Dravo Corp., Pittsburgh, Pa. 1940; Design Engineer, Clearing Machine Co., Chicago, Illinois 1941; University of Oklahoma, department of Mechanics since 1941; Present title, Professor of Mechanics and Metallurgy; hold patents on laminated pressure vessel construction; member of A.S.M.E., S.A.E., I.A.S., A.G.M.A., and ΣXi; member of Committee on Theory, Industrial Instruments and Regulators Division, A.S.M.E.; Paper in A.S.M.E. Trans. October 1941, "Combustion Explosions in Pressure Vessels Protected With Rupture Disks".

JAMES O. MELTON was born in Oklahoma, 1922; Attended University of Oklahoma 1940-43 and 1946-49. B.S. in General Engineering 1947, M. Engineering 1949; Graduate Assistant in Department of Mechanics at the University of Oklahoma 1947-48; Special Instructor in Mechanics and Metallurgy, University of Oklahoma 1948 to present. Junior member of A.S.M.E.; Member of A.S.E.E., and T Ω; Member of committee on Theory, Industrial Instruments and Regulators Division, A.S.M.E. Engaged in Automatic control research.

The natural frequencies of an undamped system of masses and springs can be found by finding the roots of the following equation

$$u^n + au^{n-1} + bu^{n-2} + \ldots\ldots\ldots\ldots\ldots u + p = 0 \qquad (1)$$

where the coefficients a, b, c, etc., are all positive. The natural frequencies are then given by

$$\omega^2 = -u \qquad (2)$$

where u is a root of Equation (1) and ω is a natural frequency of the system.

When the system consists of N masses and (N − 1) springs, there will be (N − 1) natural frequencies. Equation (1) will therefore be an equation of (N − 1) degree.

In Appendix A, a simple method of finding Equation (1) is given. Since all the roots of Equation (1) are negative it is a simple matter to find them by Graeffe's root-squaring method. In Appendix B is shown the method used to find the roots of Equation (1).

To illustrate the ease with which the natural frequencies can be found an example will be worked.

Numerical Example.

FIGURE 1.—Equivalent Five-Mass, Four-Spring System.

Suppose we have a four cylinder engine and flywheel. It is first reduced to an equivalent 5-mass, 4-spring system, as shown in Figure 1, by methods well covered in the literature. See reference (1), (2), (3), (4).

Given:

$M_1 = 10$ slugs ft.^{2}

$M_2 = M_3 = M_4 = M_5 = 0.1$ slugs ft.^{2}

$K_1 = 1.3 \times 10^6$ ft. lb./radian.

also by Equation (9) Appendix A

$a_1 = 13 \times 10^6$

$a_1 = a_3 = a_4 = 15 \times 10^6$

and by Equation (10) Appendix A

$b_1 = 0.13 \times 10^6$

$b_2 = b_3 = b_4 = 15 \times 10^6$

also

$a_0 = 0, \quad F_0 = 0, \quad$ and $F_5 = 0$

Now writing Equation (12), Appendix A, for F_0, F_1 and F_2 and solving for F_2:

$$F_2 = \frac{u}{a_1} F_1 + \frac{a_1 + b_1}{a_1} F_1 \tag{3}$$

And writing Equation (16) Appendix A, for F_1, F_2 and F_3 and solving for F_3

$$F_3 = \frac{u}{a_2} F_2 + \frac{a_2 + b_2}{a_2} F_2 - \frac{b_2}{a_2} F_1 \tag{4}$$

and writing Equation (12) Appendix A, for F_2, F_3 and F_4 and solving for F_4.

$$F_4 = \frac{u}{a_3} F_3 + \frac{a_3 + b_3}{a_3} F_3 - \frac{b_3}{a_3} F_2 \tag{5}$$

also writing Equation (12) Appendix A, for F_3, F_4 and F_5 and solving for F_5, which is equal to zero:

$$F_5 = \frac{u}{a_4} F_4 + \frac{a_4 + b_4}{a_4} F_4 - \frac{b_4}{a_4} F_3 = 0 \tag{6}$$

The variables F_1, F_2, F_3 and F_4 can be eliminated from Equations (3), (4), (5) and (6) leaving a 4th degree equation in u and the coefficients a and b.

This can best be done by means of a tabulation using the numerical values of the coefficients a and b. The equations then become:

$$F_2 = \frac{u}{13 \times 10^6} F_1 + \frac{13.13 \times 10^6}{13 \times 10^6} F_1 \tag{3a}$$

$$F_3 = \frac{u}{15 \times 10^6} F_2 + \frac{3.0 \times 10^6}{15 \times 10^6} F_2 - \frac{15 \times 10^6}{15 \times 10^6} F_1 \tag{4a}$$

$$F_4 = \frac{u}{15 \times 10^6} F_3 + \frac{30 \times 10^6}{15 \times 10^6} F_3 - \frac{15 \times 10^6}{15 \times 10^6} F_2 \tag{5a}$$

$$0 = \frac{u}{15 \times 10^6} F_4 + \frac{30 \times 10^6}{15 \times 10^6} F_4 - \frac{15 \times 10^6}{15 \times 10^6} F_3 \tag{6a}$$

These are tabulated on insert number 1.

The tabulation gives the following fourth degree equation:

$$u^4 \times 10^{-24} + 103.44499\, u^3 \times 10^{-18} + 3246.1499\, u^2 \times 10^{-12} + 31342.50107\, u \times 10^{-6} + 45630.0015 = 0$$

but substituting

$$u = u_1 \times 10^6$$

in the above:

$$u_1^4 + 103.44499\, u_1^3 + 3246.1499\, u_1^2 + 31342.50107\, u_1 + 45630.0015 = 0$$

which will now be solved by the method developed in Appendix B. (See insert number 2.)

Therefore:

$$u_1 = -52.643$$
$$u_1 = -34.383$$
$$u_1 = -14.347$$
$$u_1 = -1.757$$

But since $-u = \omega^2$ the natural frequencies of this five mass — four spring system are:

$$\omega_1 = \sqrt{52.643 \times 10^6} = 7.255 \times 10^3 \text{ rad./sec.}$$
$$\omega_2 = \sqrt{34.383 \times 10^6} = 5.863 \times 10^3 \text{ rad./sec.}$$
$$\omega_3 = \sqrt{14.347 \times 10^6} = 3.788 \times 10^3 \text{ rad./sec.}$$
$$\omega_4 = \sqrt{1.757 \times 10^6} = 1.302 \times 10^3 \text{ rad./sec.}$$

The foregoing is offered as a rather straightforward method of determining the natural frequencies. It is the authors' opinion that the natural frequencies can be found by this method with less labor than by the Holzer method. Every thing used here is available in the existing literature. However, the method of finding the frequency equation in combination with the Graeffe Root squaring method is here used for the first time as far as the authors are aware.

Appendix A.

Suppose the mass M in Fig. 1A, is sliding back and forth, without friction in a straight line on a smooth plane, with simple harmonic motion. When the mass is at rest and displaced a distance x from its position of static equilibrium, we can write,

$$-kx = MA$$

but

$$A = -x\omega^2 \qquad (1)$$

Then the relation between the harmonic force F applied to the mass and the angular frequency of its motion is

$$F = xM\omega^2 \qquad (2)$$

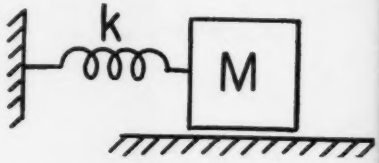


FIGURE 1A.

Next consider several masses connected by springs, as shown in Fig. 2A.

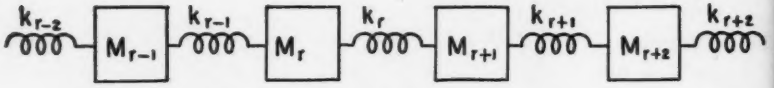


FIGURE 2A.

Now let any mass M_r to be displaced to the right a distance x_r then from Equation (2) we can write

$$(F_{r-1} - F_r) = x_r M_r \omega^2 \qquad (3)$$

and for Mass M_{r+1}

$$F_r - F_{r+1} = M_{r+1}\, x_{r+1}\omega^2 \qquad (4)$$

solving Equations (3) and (4) for x_r and x_{r+1}

$$x_r = \frac{F_{r-1} - F_r}{M_r\omega^2} \qquad (5)$$

$$x_{r+1} = \frac{F_r - F_{r+1}}{M_{r+1}\omega^2} \qquad (6)$$

but we have

$$F_r = k_r\,(x_{r+1} - x_r) \qquad (7)$$

substituting Equations (5), and (6) in (7)

$$F_r\omega^2 = k_r\left(\frac{F_r - F_{r+1}}{M_{r+1}} - \frac{F_{r-1} - F_r}{M_r}\right) \qquad (8)$$

but write

$$\frac{k_r}{M_{r+1}} = a_r \qquad (9)$$

$$\frac{k_r}{M_r} = b_r \qquad (10)$$

tion in a
mass is
an write,

Equation

(3)

(4)

(5)

(6)

(7)

(8)

(9)

(10)

	F_1u^6	
$F_2 = \frac{a_1 + b_1}{a_1} F_1 + \frac{u}{a_1} F_1$		
$\frac{a_1 + b_1}{a_1} F_1 =$		
$\frac{u}{a_1} F_1 =$	1.01	
$F_2 =$	1.01	
$F_3 = -\frac{b_2}{a_2} F_1 + \frac{a_2 + b_2}{a_2} F_2 + \frac{u}{a_2} F_2$		
$-\frac{b_2}{a_2} F_1 =$	−1.00	
$\frac{a_2 + b_2}{a_2} F_2 =$	2.02	
$\frac{u}{a_2} F_2 =$		
$F_3 =$	1.02	
$F_4 = -\frac{b_3}{a_3} F_2 + \frac{a_3 + b_3}{a_3} F_3 + \frac{u}{a_3} F_3$		
$-\frac{b_3}{a_3} F_2 =$	−1.01	−(
$\frac{a_3 + b_3}{a_3} F_3 =$	2.04	(
$\frac{u}{a_3} F_3 =$		(
$F_4 =$	1.03	(
$F_5 = -\frac{b_4}{a_4} F_3 + \frac{a_4 + b_4}{a_4} F_4 + \frac{u}{a_4} F_4$		
$-\frac{b_4}{a_4} F_3 =$	−1.02	−(
$\frac{a_4 + b_4}{a_4} F_4 =$	2.06	(
$\frac{u}{a_4} F_4 =$		(
$F_5 =$	1.04	(

Dividing through by F_1 and the coefficient of u^4 gives

Since $F_5 = 0$:	$0 =$	45630.001

NOTE: Headings of the various columns are multiplie

F_1u^0	$F_1u^1 \times 10^{-6}$	$F_1u^2 \times 10^{-12}$	$F_1u^3 \times 10^{-18}$	$F_1u^4 \times 10^{-24}$
	0.0769230769			
1.01				
1.01	0.0769230769			
−1.00				
2.02	0.1538461538			
	0.0673333333	0.005128205		
1.02	0.2211794871	0.005128205		
−1.01	−0.0767230796			
2.04	0.4423589742	0.01036410		
	0.0680000000	0.014745299	0.00034188033	
1.03	0.4334358973	0.025109399	0.00034188033	
−1.02	−0.2211794871	−0.005128205		
2.06	0.8668717946	0.050218798	0.00068376066	
	0.0686666667	0.02889572648	0.001673959933	0.000022792022
1.04	0.7143589741	0.07398631948	0.002357720593	0.000022792022

ient of u^4 gives

45630.001	31342.50107	3246.1499	103.44499	1

nns are multiplied by the numerical coefficients tabulated in the table.

INSERT NUMBER 1

m	u_1^4	u_1^3
1	1	103.13
	1	10635.7969
		−6473.40
2	1	4162.3969
	1	17.325547953×10^6
		-8.205565680×10^6
4	1	9.11998227×10^8
	1	$83.174076605 \times 10^{12}$
		$-22.23619754 \times 10^{12}$
8	1	$60.93787907 \times 10^{12}$
	1	$37.134251055 \times 10^{28}$
		$-2.306315642 \times 10^{28}$
16	1	$34.82793542 \times 10^{28}$
	1	$12.1298508561 \times 10^{54}$
		$-0.026545055 \times 10^{54}$
32	1	$12.10330580 \times 10^{54}$

u_1^3	u_1^2	u_1^1	u_1^0
	3236.7	31342.5	45630
9	$10.47622689 \times 10^{6}$	9.82352306×10^{8}	20.8209690×10^{8}
	$-6.46470405 \times 10^{6}$	$-2.95381242 \times 10^{8}$	
	$+0.091260 \times 10^{6}$		
	4.10278284×10^{6}	6.86971064×10^{8}	20.8209690×10^{8}
953×10^{6}	$16.832827032 \times 10^{12}$	$47.192924277 \times 10^{16}$	$4.33512750 \times 10^{18}$
80×10^{6}	$-5.718892454 \times 10^{12}$	$-1.7084782865 \times 10^{16}$	
	$+0.0041641938 \times 10^{12}$		
7×10^{6}	$11.11809877 \times 10^{12}$	$45.48444599 \times 10^{16}$	$4.33512750 \times 10^{18}$
605×10^{12}	$123.61212026 \times 10^{24}$	$20.68834827 \times 10^{34}$	$18.79333044 \times 10^{36}$
54×10^{12}	$-8.2963468198 \times 10^{24}$	$-0.009639675 \times 10^{34}$	
	$+0.0000086703 \times 10^{24}$		
07×10^{12}	$115.31578211 \times 10^{24}$	$20.67870859 \times 10^{34}$	$18.79333044 \times 10^{36}$
055×10^{36}	$1.3297729601 \times 10^{52}$	$4.2760898895 \times 10^{70}$	$3.5318926902 \times 10^{74}$
42×10^{36}	$-0.0025202333 \times 10^{52}$	$-0.0000000433 \times 10^{70}$	
	$+0.3758666088 \times 10^{52}$		
42×10^{36}	$1.3272527268 \times 10^{52}$	$4.2760898462 \times 10^{70}$	$3.5318926902 \times 10^{74}$
8561×10^{54}	$1.7615998013 \times 10^{104}$	$18.284944371 \times 10^{140}$	$12.4742659737 \times 10^{148}$
55×10^{54}	$-0.000002978547 \times 10^{104}$	$-93.7542840 \times 10^{135}$	
	$+0.7063785380 \times 10^{75}$		
80×10^{54}	$1.7615968228 \times 10^{104}$	$18.28494437 \times 10^{140}$	$12.4742659737 \times 10^{148}$

INSERT NUMBER 2

then

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$F_1\left(u\right.$

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$$F_r\omega^2 = a_r(F_r - F_{r+1}) - b_r(F_{r-1} - F_r) \tag{11}$$

from Equation (11)

$$F_r(-\omega^2 + a_r + b_r) = a_r F_{r+1} + b_r F_{r-1} \tag{12}$$

By writing Equation (12) for as many times as there are springs we have a set of equations in the unknowns F and ω^2. From this set, F can be eliminated, and an equation in ω^2 and the elastic constants a and b can be written.

For example consider the simple system, shown in Fig. 3A.

then

$$a_1 = \frac{k_1}{M_2}$$

$$b_1 = \frac{k_1}{M_1}$$

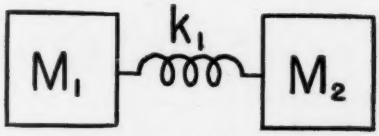

FIGURE 3A.—Equivalent Two-Mass, One-Spring System.

let $-u = \omega^2$

then Equation (12) becomes

$$F_1\left(u + \frac{k_1}{M_2} + \frac{k_1}{M_1}\right) = \frac{k_1}{M_2} F_2 + \frac{k_1}{M_1} F_0$$

but $F_0 = 0$, since there is no spring k_0.
and $F_2 = 0$, since there is no spring k_2.

then

$$F_1\left(u + \frac{k_1}{M_2} + \frac{k_1}{M_1}\right) = 0$$

$$u = -\frac{k_1 (M_1 + M_2)}{M_1 M_2}$$

but $-u = \omega^2$
therefore

$$\omega = \sqrt{\frac{k_1 (M_1 + M_2)}{M_1 M_2}}$$

which is the natural frequency for a two mass, single spring system.

Next consider a system consisting of three masses and two springs. (Fig. 4A.) then

$$a_1 = \frac{k_1}{M_2}$$

$$a_2 = \frac{k_2}{M_3}$$

$$b_1 = \frac{k_1}{M_1}$$

$$b_2 = \frac{k_2}{M_2}$$

FIGURE 4A.—Equivalent Three-Mass, Two-Spring System.

Substituting in Equation (12), letting $-u = \omega^2$

$$F_1(u + a_1 + b_1) = a_1 F_2 + b_2 F_0$$
$$F_2(u + a_2 + b_2) = a_2 F_3 + b_2 F_1$$

but since there are no springs k_0 or k_3

$$F_0 = 0 \quad \text{and} \quad F_3 = 0$$

therefore

$$F_1(u + a_1 + b_1) = a_1 F_2$$

and

$$F_2(u + a_2 + b_2) = b_2 F_1$$

Eliminating F_2

$$F_1\left(\frac{u + a_1 + b_1}{a_1}\right) = F_1\left(\frac{b_2}{u + a_2 + b_2}\right)$$

then

$$u^2 + u\left(\frac{k_1}{M_1} + \frac{k_1}{M_2} + \frac{k_2}{M_2} + \frac{k_2}{M_3}\right) + \frac{k_1k_2}{M_1M_2} + \frac{k_1k_2}{M_2M_3} + \frac{k_1k_2}{M_1M_3} = 0$$

but $-u = \omega^2$

therefore

$$\omega^4 - \omega^2\left(\frac{k_1[M_1M_3 + M_2M_3] + k_2[M_1M_2 + M_1M_3]}{M_1M_2M_3}\right) + \left(\frac{k_1k_2M_1 + k_1k_2M_2 + k_1k_2M_3}{M_1M_2M_3}\right) = 0$$

In any system consisting of more than three masses it will be much more convenient to solve the set of equations by means of the following tabulation.

	F_1u^0	F_1u^1	F_1u^2	F_1u^3
$F_2 = \frac{a_1 + b_1}{a_1} F_1 + \frac{uF_1}{a_1}$				
$\frac{a_1 + b_1}{a_1} F_1 =$	$\frac{a_1 + b_1}{a_1}$			
$\frac{uF_1}{a_1} =$		$\frac{1}{a_1}$		
$F_2 =$	$\frac{a_1 + b_1}{a_1}$	$\frac{1}{a_1}$		
$F_3 = -\frac{b_2}{a_2} F_1 + \frac{a_2 + b_2}{a_2} F_2 + \frac{uF_2}{a_2}$				
$-\frac{b_2}{a_2} F_1 =$	$\frac{-b_2}{a_2}$			
$\frac{a_2 + b_2}{a_2} F_2 =$	$\left(\frac{a_2 + b_2}{a_2}\right)\left(\frac{a_1 + b_1}{a_1}\right)$	$\left(\frac{a_2 + b_2}{a_2}\right)\frac{1}{a_1}$		
$\frac{uF_2}{a_2} =$		$\frac{a_1 + b_1}{a_1a_2}$	$\frac{1}{a_1a_2}$	
$F_3 =$	Algebraic sum of terms up to double line.			
$F_4 =$				

Repeat as often as required.

APPENDIX B.

Numerical Solution of Higher Degree Equations.

The following is Graeffe's Root-Squaring method of finding the roots of a higher degree equation. Suppose we have any equation with real positive coefficients:

$$x^n + A_1x^{n-1} + A_2x^{n-2} + \cdots + A_n = 0$$

The coefficients of this equation are tabulated as follows:

m	1	A_1	A_2	A_3	A_4	A_5	....	A_n
1	1	$(A_1)^2$ $-2A_2$	$(A_2)^2$ $-2A_1A_3$ $+2A_4$	$(A_3)^2$ $-2A_2A_4$ $+2A_1A_5$ $-2A_6$	$(A_4)^2$ $-2A_3A_5$ $+2A_2A_6$ $-2A_1A_7$ $+2A_8$	$(A_5)^2$ $-2A_4A_6$ $+2A_3A_7$ $-2A_2A_8$	....	$(A_n)^2$
2	1	$(A_1)^2-2A_2$	(Algebraic sum of numbers up to double line)					

$B_1 = (A_1)^2 - 2A_2$

$B_2 = (A_2)^2 - 2A_1A_3 + 2A_4$

$B_3 = (A_3)^2 - 2A_2A_4 + 2A_1A_5 - 2A_6$

$B_4 = (A_4)^2 - 2A_3A_5 + 2A_2A_6 - 2A_1A_7 + \cdots\cdots$

$B_5 = (A_5)^2 - 2A_4A_6 + 2A_3A_7 - 2A_2A_8 + 2A_1A_9 - \cdots\cdots$

2	1	B_1	B_2	B_3	B_4	B_5	....	B_n
		$(B_1)^2$ $-2B_2$	$(B_2)^2$ $-2B_1B_3$ $+2B_4$	$(B_3)^2$ $-2B_2B_4$ $+2B_1B_5$ $-2B_6$	$(B_4)^2$ $-2B_3B_5$ $+2B_2B_6$ $-2B_1B_7$ $+2B_8$	$(B_5)^2$ $-2B_4B_6$ $+2B_3B_7$ $-2B_2B_8$	....	$(B_n)^2$
4	1	$(B_1)^2-2B_2$	(Algebraic sum of numbers up to double line)					

$C_1 = (B_1)^2 - 2B_2$

$C_2 = (B_2)^2 - 2B_1B_3 + 2B_4$

$C_3 = (B_3)^2 - 2B_2B_4 + 2B_1B_5 - 2B_6$

$C_4 = (B_4)^2 - 2B_3B_5 + 2B_2B_6 - 2B_1B_7 + 2B_8 + \text{etc.}$

$C_5 = (B_5)^2 - 2B_4B_6 + 2B_3B_7 - 2B_2B_8 + \text{etc.}$

This tabulation is continued until the sum of every column is equal to the squared term at the beginning of the column to any required number of significant figures.

The Roots of the original equation will be represented by $x_1, x_2, x_3 \ldots\ldots x$.

$$\log x_1 = (1/m)\ (\log C_1)$$

$$\log x_2 = (1/m)\ (\log C_2 - \log C_1)$$

$$\log x_3 = (1/m)\ (\log C_3 - \log C_2)$$

$$\log x_n = (1/m)\ (\log C_n - \log C_{n-1})$$

EXAMPLE: Find the roots of the following Equation:

$$x^4 + 149\,x^3 + 6211\,x^2 + 73419\,x + 203580 = 0$$

(See insert number 3)

$$\log x_1 = \frac{\log 116.041441815 \times 10^{60}}{32}$$

$$\log x_1 = \frac{62.0646077}{32} = 1.9395190$$

$$x_1 = 87$$

$$\log x_2 = \frac{\log 9.277191156 \times 10^{114} - \log 116.04144188 \times 10^{60}}{32}$$

$$\log x_2 = \frac{114.9674169 - 62.0646077}{32} = 1.6532127$$

$$x_2 = 45$$

likewise:

$$x_3 = 13$$

$$x_4 = 4$$

Therefore the roots of the equation $x^4 + 149x^3 + 6211x^2 + 73419x + 203580 = 0$ are 87, 45, 13, and 4.

REFERENCES.

1. Wilson, W. Ker—"Practical Solution of Torsional Vibration Problems," John Wiley and Son, Inc., New York, 1942.

2. Timoshenko, S.—"Vibration Problems in Engineering," D. Van Nostrand Company Inc., New York, 1937.

3. Den Hartog, J. P.—"Mechanical Vibrations," McGraw-Hill Book Company Inc., New York, 1947.

4. Tuplin, W. A.—"Torsional Vibrations," John Wiley and Sons, New York, 1943.

5. Doherty, R. E., and Keller, E. G.—"Mathematics of Modern Engineering," Vol. 1, John Wiley and Sons, Inc., New York, 1936.

6. Whitaker, E. G., and Robinson, C.—"Calculus of Observations," Blackie and Son Ltd., London, 1924.

7. Scarborough, J. B.—"Numerical Mathematical Analysis," The Johns Hopkins Press, Baltimore, 1930.

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m	x^4	x^3
1	+1	+149
	+1	22201
		−12422
2	1	9779
	1	95.628841×10^6
		-34.209638×10^6
4	1	61.419203×10^6
	1	$3772.3184972 \times 10^{12}$
		$-473.385404 \times 10^{12}$
8	1	$3298.932657 \times 10^{12}$
	1	$10.882956675421 \times 10^{20}$
		$-0.11038379068 \times 10^{20}$
16	1	$10.7725728847 \times 10^{20}$
	1	$116.0483264549 \times 10^{40}$
		$-0.0060916965 \times 10^{40}$
32	1	$116.041441815 \times 10^{40}$

x^3	x^2	x^1	x^0
	+6211	+73419	+203580
	38576521	5390349561	41444816400
	−21878862	−2528870760	
	407160		
	17104819	2861478801	41444816400
$\times 10^{6}$	292574833022761	$8.188060928572398 \times 10^{18}$	$17.176728064297 \times 10^{20}$
$\times 10^{6}$	−55964802389958	$-1.4178121660204632 \times 10^{18}$	
	82889632800		
$\times 10^{6}$	$236692920.2 \times 10^{6}$	$6.770248763 \times 10^{18}$	$17.176728064297 \times 10^{20}$
972×10^{12}	$56023.5384728 \times 10^{24}$	$45.83626831 \times 10^{36}$	$2.9503998685 \times 10^{42}$
$\times 10^{12}$	$-831.6465665 \times 10^{24}$	$-0.8131219848 \times 10^{36}$	
	$0.0034353456 \times 10^{24}$		
7×10^{12}	$55191.89534 \times 10^{24}$	$45.0231463281 \times 10^{36}$	$2.9503998685 \times 10^{42}$
575421×10^{30}	$3046.14531122 \times 10^{54}$	$2027.0837054526 \times 10^{72}$	$8.70485938109 \times 10^{84}$
9068×10^{30}	$-0.2970566555 \times 10^{54}$	$-0.3256763215 \times 10^{72}$	
	$5.900799736 \times 10^{42}$		
3847×10^{30}	$3045.848254565 \times 10^{54}$	$2026.758029131 \times 10^{72}$	$8.70485938109 \times 10^{84}$
4549×10^{60}	$9.27719159249 \times 10^{114}$	$4.107748108 \times 10^{150}$	$75.774576843 \times 10^{168}$
765×10^{60}	$-0.043666792 \times 10^{108}$	$-0.000000053014 \times 10^{150}$	
	$17.40971876 \times 10^{84}$		
815×10^{60}	$9.277191156 \times 10^{114}$	$4.107748055 \times 10^{150}$	$75.774576843 \times 10^{168}$

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LONGITUDINAL VIBRATION—An energy method of estimating frequency.

BY C. S. L. ROBINSON.

THE AUTHOR.

MR. ROBINSON was graduated from Harvard University A.B. cum laude and M.S. in mechanical engineering. From 1935 until 1941 he was employed in several apparatus departments of the General Electric Company after taking a year of their engineering advanced course.

Since 1941 he has been with the Shipbuilding Division of Bethlehem Steel Company, working on design problems in marine engineering and doing developmental work in vibrations and fluid dynamics.

The Rayleigh method is used here to solve for resonant speeds excited by propeller blade frequency thrust impulses. These are important in large geared turbine ships with long shafting.

The use of an energy consideration rather than a system of force equations (as in the Holzer method) is made desirable by the fact that the mass of the shafting is considerable. The fundamental relationship is:

Max Total Kinetic Energy = Max Total Potential Energy

This method is used to solve for the lowest natural frequency, which is the only one low enough to be considered. An adequate shape for the curve of vibration amplitudes can be calculated by considering the shafting with attached masses as a column loaded in compression or tension. Actual amplitudes are not absolutely determined because excitation and damping are not included.

Nomenclature:

All flexibilities are expressed as equivalent length of line shaft.

A = Cross section area of line shaft.
f = Deflection or assumed amplitude at one point.
f_{av} = Average deflection between two points.
Δf = Relative deflection between two points.
W = Weight of a rigid mass at one point.
$w\Delta x$ = Weight of a length of shafting between two points.
F = Axial force in shafting at one point.
F_{av} = Average axial force in shafting between two points.
ω = Natural frequency in radians per second.

I Deflections:

Refer to Figure 1

The node will be located where the foundation flexibility begins. This may be considered the anchor point, and the weights of the vibrating masses may be considered the loads. All flexibilities including that of the thrust bearing should be converted to equivalent length of line shafting.

A curve of total load (F) vs. equivalent length should be constructed. Concentrated masses will be represented by abrupt increases in F while the distributed mass of the shafting will be represented by a gradual increase.

The loading, (F) divided by AE is the slope of the deflection curve. Hence the deflection curve itself may be found by numerical integration The anchor is taken as the point of zero deflection.

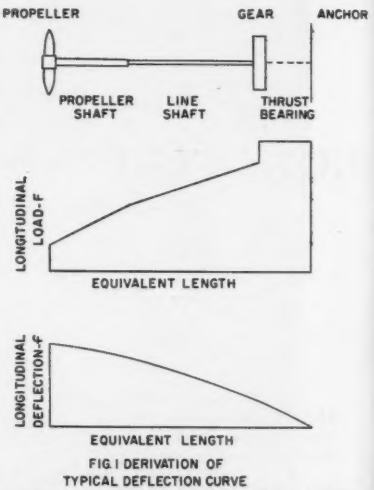


FIG. I DERIVATION OF TYPICAL DEFLECTION CURVE

II Kinetic Energy:

The maximum vibration kinetic energy is

$$\frac{\omega^2}{2g}\left(\Sigma Wf^2 + \Sigma wax\ f\frac{2}{av}\right)$$

The summation can be made from a tabulation of the known loads and the deflections calculated above, with ω^2 remaining unknown.

III Potential Energy:

The maximum vibration potential energy is

$$1/2\ \Sigma\ F_{av}\ \triangle\ f$$

It will be apparent that this is determined from a tabulation of the strain energy in each section of (equivalent) shafting.

IV Natural Frequency:

From equating the kinetic energy to the potential energy

$$\omega^2 = g\frac{\Sigma\ Fav\ \triangle\ f}{(\Sigma\ Wf^2 + \Sigma\ wax\ f_{av}^2)}$$

This is for the one node or lowest natural frequency. To date higher nodes have frequencies so far beyond any that can be excited by the propeller that their calculation is not necessary.

Conclusion:

The novelty of this method lies in its consideration of the shafting as distributed mass rather than lumping it with the gear and the propeller. This refinement is of value since the shafting weight may be several times that of any other component. This is true with long shafting and it is only with long shafting that primary resonance will be encountered. In torsional vibration problems the small radius of gyration of the shafting makes its rotational inertia negligible. A corresponding analogy does not apply to longitudinal vibration.

Another advantage of this method is that it could include the effect of adjacent masses in the machinery room if they should be coupled to the vibrating system. After their relative amplitudes are estimated or observed, their influence on the resonant frequency may be surmised by including their kinetic and potential energies in the calculations.

REFERENCES.

J. P. DenHartog "Mechanical Vibrations" 3rd Edition McGraw Hill Book Co. 1947.

C. P. Rigby "Longitudinal Vibration of Marine Propeller Shafting"—noted in JOURNAL OF AMERICAN SOCIETY OF NAVAL ENGINEERS, P341, August 1948.

A CONTROLLABLE PROPELLER SYSTEM—A Design Proposed as Adaptable to Naval Boats and Small Craft.

BY HARRY J. NICHOLS.

THE AUTHOR.

Attended U. S. Naval Academy 1913-1916. Served in U. S. Naval Reserve on active duty through World War I. After 1920, was Engineer-in-Charge of the Experimental Ammunition Unit of the Naval Ordnance Laboratory then located at the Naval Gun Factory, Washington, D. C. In 1929 left government employment and for twelve years was employed in engineering and development work by private industry. From 1941 to 1946 was a Consulting Development Engineer, New York City. Presently is President of Amplitorque. Has to his credit more than 100 inventions with more than 70 patented and about 30 now pending.

INTRODUCTION.

It is the main object of this paper to describe a controllable propeller system for boats and small craft considered by the writer to be suitable for naval use, and to point out some of the advantages to be gained thereby.

Before we can hope to understand and evaluate the solution of a problem, we must first understand the problem itself. In order that this proposed solution of a complex propeller problem may be understood by the general readers of this paper, as well as by propeller experts, no assumption of prior knowledge of the subject will be made. It is hoped those who have such knowledge will bear with the preliminary and elementary portions in order that a common starting point may be established.

For present purposes, a controllable propeller system may be defined as a marine propulsion system whereby the angle of the propeller blades can be controlled as desired by a master control member within the vessel. Such a system involves more than merely equipping a vessel with a different type of propeller, but actually amounts to providing a vessel with a distinct type of coordinated propulsion system which substantially improves the over-all performance of the engine and the vessel itself.

Since the general trend is towards the use of Diesel engines for the propulsion of boats and small craft, it will be assumed that the vessel is equipped or will be equipped with one or more Diesel engines. It also will be assumed that each engine is equipped with a throttle control or governor control, whereby the speed of the engine can be regulated as desired.

A controllable propeller system can perform the six-fold function of:

(1) Providing a neutral position of the propeller blades whereby the engine can be started and run idly without propelling the vessel,

(2) Precisely controlling the motion of the vessel when maneuvering in close quarters,

(3) Reversing the direction of the propeller thrust without reversing the direction of rotation of the propeller thereby stopping the vessel or reversing its direction,

(4) Providing a variable ratio between the propeller and engine torques thereby permitting full engine power to be used under heavy load conditions,

(5) Providing a precise blade adjustment whereby the blades can be adjusted for maximum propeller performance under any operating conditions, and

(6) Providing over-pitch to obtain maximum fuel economy at cruising speeds. These functions are analogous in the main to the combined functions of the neutral shift, clutch, brakes, reverse gears, variable ratio gearing or torque converter, and over-drive provided in some recent models of automobiles, trucks and busses.

A controllable propeller system can provide a degree of flexibility in the application of Diesel engines to marine propulsion which has heretofore only been achieved by electric drive, but it is inherently more efficient and less costly than electric-drive. Moreover, when equipped with a controllable propeller system, a vessel can be handled with the ease, precision, and certainty of the modern automobile, its performance and operating efficiency can be substantially improved, and navigational safety increased.

GENERAL CONSIDERATIONS.

In the design of controllable propeller systems, as in most marine equipment, it is considered that primary emphasis should be placed on reliability, simplicity, ruggedness, compactness, ease and flexibility of installation, minimum maintenance, accessibility, long life and low cost. These design objectives are most readily achieved by reducing the mechanism carried by the propeller shaft to a minimum, and by utilizing the maximum number of commercially available standard components in the design.

A further major consideration for the proposed field of application is that the controllable propeller system should be versatile and adaptable to the wide range of engines, engine combinations, shaft sizes and propeller sizes used in the boats, small craft and landing craft of the Navy. To this end, the system should be such that it can be designed to be separate from the engine if desired, it should not require special shafting, and the accessory apparatus should be suitable for locating wherever most convenient.

A further desirable consideration is that the system should be one of calculable performance which can be designed on an engineering basis to meet specific requirements, and adapted to the development of a line of equipment of the same general design, but of graduated sizes, with a view to eventual standardization. Moreover, the system should be adaptable to the replacement of fixed-blade propellers in existing propulsion installations, for purposes of increasing the available power and improving the performance of such installations.

PROPELLER SYSTEM.

DESIGN REQUIREMENTS.

On study of the design requirements in detail, under the above stated premises, it appeared that the main problems were to design a compact, rugged and streamlined hub assembly of minimum feasible diameter which would enable the propeller to be mounted in the usual location and manner; to devise simple and powerful blade actuating mechanism mounted entirely within the hub; to provide operating connections for that mechanism which will enable a propeller shaft of normal diameter and construction to be used; to provide suitable and reliable power means for driving the blade actuating mechanism; and to provide a flexible pitch control system and an accurate pitch indicating system both adapted to remote control.

GENERAL DESCRIPTION OF THE SYSTEM.

The result of the study is a system which comprises generally a hub assembly including a hydraulically actuated mechanical movement, for turning the blades, axially mounted entirely in the hub; a hollow propeller shaft of normal size and construction having a single fixed conduit of relatively small size mounted coaxially therein, a hydraulic fluid transfer unit mounted around the propeller shaft within the vessel; hydraulic apparatus located conveniently anywhere in the engine room comprising a tank, pump, and a master cylinder; hydraulic fluid control valves comprising a relief valve, a selector-valve operatively connected to the master cylinder, and a fluid locking valve for normally maintaining fluid pressure in the master and hub cylinders, thereby to hold the blades at any set pitch; combined pitch control and pitch indicating means including a master control member operatively connected to the selector-valve, and semi-automatic means for synchronizing the movements of the blades with those of the master control member. This system as a whole constitutes a hydraulic servo-system of position control type.

In the following description, the parts particular to this propeller system are designated by numerals or letters; the usual parts, including commercial fastenings, being referred to by name only.

THE PROPELLER ASSEMBLY.

Referring now to Fig. I, the propeller assembly consists of a one-piece hub H, having a blind axial bore and open radial blade sockets, bolted tightly to a flange on the outboard end of the hollow propeller shaft so as to provide a closed chamber; blades B mounted rotatably in the blade sockets; and a hydraulically actuated mechanical movement, for turning the blades axially in unison so as to vary their pitch, mounted entirely with the hub.

Because of the powerful forces to which a blade may be subjected, particularly during crash-stop conditions or when the blades may strike a heavy obstruction, the design of the blade mounting is a vital factor in all variable pitch propellers. The blade mounting here shown was evolved after study of various blade mountings which have been used successfully on variable pitch hydraulic, marine and airplane propellers. The major normal force

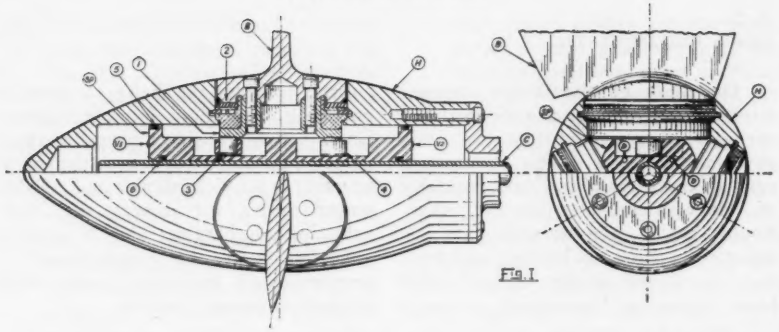


Fig. I.

acting on a propeller blade in operation is centrifugal force, and the moment of this force referred to the thrust bearings is counter to the moment of the thrust and torque. The resultant of all forces normally acting on a blade is directed radially outward in the main, hence the heaviest load comes on the inner thrust bearing. However, the main object in the design of the blade mounting shown is to absorb heavy impact forces without damage.

The blades are each provided with a circular boss and are mounted for rotation in the blade sockets by means of a demountable flanged journal (1), detachably secured to the boss by cap screws, and a demountable blade mounting (2) here comprising a combination sleeve bearing, two-way ring thrust bearings and fluid seal. The blade mounting consists of a bearing ring held rigidly in place by means of a "Truarc" beveled retainer ring, a spring washer, and a plastic thrust washer, as shown. While the retainer ring may appear small in cross-section, its actual holding strength is several times that of the blade bolts. The plastic thrust washer and spring washer enable the thrust bearings to be preloaded to eliminate shake without excessive starting friction or undue wear. Such bearing friction as results from preloading and the operating load acts to dampen out any tendency towards blade vibration. The working bearing loads are well within the limits established by practice for the bearing materials and lubrication provided. Moreover, the design shown provides a greater bearing span and greater projected bearing areas than any so-called straddle bearings which can be designed to go into a hub of the same relative diameter. These facts make it both practical and economical to use a single sleeve bearing in combination with ring thrust bearings, as shown.

Each blade mounting is sealed against entrance of water or loss of lubricant by an O-ring packing, as shown, as well as by the plastic thrust washer.

Each journal is provided with a pair of diametral crank-pins (3), each crank-pin being fitted with a roller sleeve (4), or a slipper block for large propellers, to minimize friction and wear. The mechanical movement for turning the blades in their bearings consists of these crank-pins and a combined wedge-and-piston, here functioning as a servo-piston (SP), mounted coaxially slidable in the bore of the hub. The piston has symmetrical lateral faces, one for each blade; each lateral face having formed therein a pair of oppositely inclined curved wedge grooves (WG), (See Fig. II), in which the pair of crank-pins engaging therewith are forced to move as followers upon axial translation of the wedge. The wedge-grooves (WG) apply a *torque-couple* to the blades by

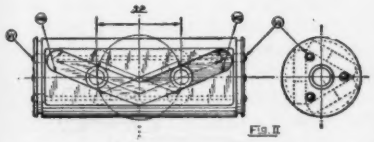


Fig. II

means of the pair of diametral crank-pins upon axial movement of the servo-piston, and are of special curvature so as to cause uniform angular displacement of the blades in direct proportion to such movement. This mechanical movement also provides a substantial mechanical advantage due to wedge action, and this mechanical advantage, combined with torque-couple action, enables the blades to be turned with a minimum of working force and of binding action of the journals in their bearings. Since a single driver element imparts turning motion to all blades, this movement provides ultimate mechanical simplicity. Furthermore, this twin crank-pin movement is rigid, has negligible back-lash, and is twice as strong as the conventional single crank-pin movement. The blade mountings, bearings, and blade turning mechanism are therefore well adapted to withstand heavy impact forces should the blades strike an obstruction. This construction thus contributes importantly to the ruggedness, compactness and high efficiency of the hub assembly.

A suitable supply of lubricant for the mechanical movement and the blade bearings is provided in the space between the piston and the blade mountings. This supply of lubricant does not have to be renewed or replaced except when the hub assembly is disassembled.

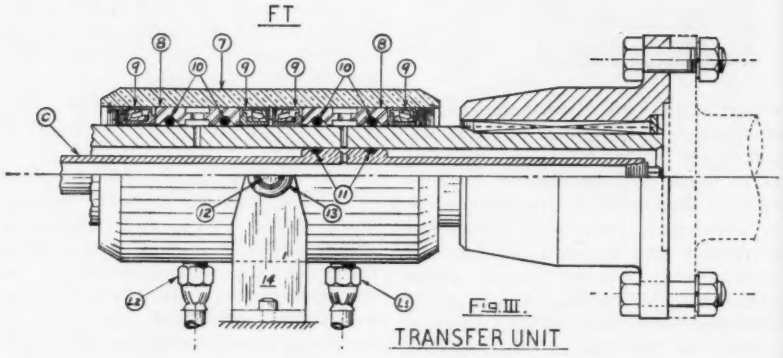
The circular ends of the servo-piston (SP) are provided with one or more annular grooves and hydraulic packings (5) of suitable type to prevent passage of fluid between the piston and the cylinder walls. The preferred form of packing is that commonly called an O-ring, which may be of any suitable composition which ensures a reliable fluid seal. The preferred hydraulic fluid is hydraulic lubricating oil, altho other suitable hydraulic fluids may be used. Oil under pressure is led to the inboard end of the piston by the hollow propeller shaft, while oil under pressure is led to the outboard end of the piston by axial conduit (C), the latter also being provided with one or more O-rings (6).

The servo-piston is also provided with one or more pairs of opposed check valves (V1), (V2) shown as spring-loaded ball check valves altho spring-loaded pin check valves may be employed instead, each pair being connected by longitudinal ducts (D) in the piston, located as shown. Each valve is assembled in the end of the piston so that the movable member of the valve, say the ball, projects slightly beyond the face of the piston, so as to be actuated by the end walls of the chamber. Thus, when the piston reaches either end of its stroke, the closed check valves at that end are pushed open, permitting oil to by-pass from the pressure to the exhaust end of the piston, thereby relieving the fluid pressure on the piston.

When oil under pressure is first introduced into the hydraulic system, the check valves by-pass any trapped air around the piston, permitting air to be eliminated from the hydraulic system without access to the hub. This same by-passing action synchronizes the stroke of the servo-piston with that of the master piston, as described hereinafter. These opposed check valves thus serve the combined functions of means for bleeding air from the hydraulic system, a pitch-limit device, and an automatic synchronizing device, but are termed generally synchronizing valves.

From the above description it will be observed that when oil under pressure is introduced into the hollow of the propeller shaft, or alternatively into conduit (C), it will act to move the servo-piston (SP), thus to change the angular position of the blades.

THE FLUID TRANSFER UNIT.



Referring now to Fig. III, which shows one form of fluid transfer unit (FT), whereby pressure oil is introduced into the propeller shaft and into conduit (C). This unit comprises generally a body (7), here having the form of a sleeve, in the bore of which two slip rings (8) and four unitary face seals (9) are assembled as shown. These face seals are preferably of differential pressure type in which the seal face is mainly pressed against the smooth edge of the adjacent slip ring by oil pressure, in well-known manner. The slip rings are held tightly but slidably on the propeller shaft by means of four O-rings (10), each mounted under compression in V-grooves formed in the bore of each ring. As the oil pressure increases, the O-rings wedge more tightly between the V-grooves and the shaft, in well-known manner. Thus the slip rings are caused to grip the shaft with increasing pressure as the oil pressure increases, preventing relative rotation, but enabling the slip rings to run true against the seal faces. The slip rings have annular grooves and radial holes, as shown, permitting passage of oil from the outer to inner peripheries thereof. The propeller shaft is provided with two sets of small radial ducts to pass fluid from the inner grooves of the respective slip rings into the interior of that shaft. Conduit (C) is provided with a tight sleeve having three V-grooves, the outer grooves being fitted with O-rings (11). Radial ducts lead from the inner groove to the interior of conduit (C), which latter is plugged tightly at the inboard end.

The slip rings (8), which are preferably of hardened steel, are a rotary fit in body (7), which latter is provided with means, such as trunnions (12) (one being shown) to prevent rotation of the transfer unit. The trunnions are preferably loosely restrained by rubber grommets (13) assembled in stationary supporting brackets (14) (one being shown) so that the transfer unit floats on the shaft, thereby enabling that unit to ride easily on and with the shaft, even if the latter runs somewhat eccentric. This is an important practical feature which avoids over-heating due to friction, and wear or scoring of the shaft; thus overcoming a main drawback which has heretofore been experienced with some propeller shaft transfer devices of the prior art. Moreover, the propeller shaft does not need to be provided with special shoulders

or collars, nor to be made of a hard metal, permitting a non-ferrous shaft to be used if preferred. External hydraulic lines (L1), (L2) conduct pressure oil from the external hydraulic apparatus into the interior of the transfer unit opposite the slip rings. This same oil lubricates all the rubbing surfaces of the transfer unit, ensuring continuous automatic lubrication, so long as oil is supplied to the unit.

PITCH CONTROL SYSTEM.

Referring now to Fig. IV, which shows a schematic diagram of the entire propeller system, a tank (T) and a pump (P) supply pressure fluid, under the control of a selector-valve (SV) which is actuated by the master control member (CM) to the propeller cylinder (PC) and a master cylinder (MC). The pump may be driven by a motor or by the engine and should have a capacity sufficient to change the pitch from full-ahead to full-astern in the specified time. The master cylinder, which is connected to impart follow-up motion to valve plunger (VP), should have a capacity substantially equal to the volume of fluid displaced during the working stroke of servo-piston (SP). A pressure actuated locking valve (LV), comprising a floating piston (FP) and opposed check valves (V3), (V4), is connected in the hydraulic circuits between the selector-valve and cylinders as shown. An adjustable safety valve (XV) by-passes the locking valve and selector-valve to relieve any abnormal pressure due to thermal expansion of the fluid, etc. While the valves can be separate units, the unitary arrangement shown simplifies the hydraulic lines. With this valve arrangement, the selector-valve may be of spring-centered, open-center type, whereby the fluid from the pump flows freely back to the tank when the pump is idling, which it does nearly all the time. This is an important practical feature of the control system which saves power and wear of the pump.

The master control member (CM) is preferably operatively associated with an indicator pointer (I) and a calibrated position scale (S), or equivalents, for purposes of indicating the pitch, as explained hereinafter. The selector-valve can be actuated by control forces of a weak order, and the

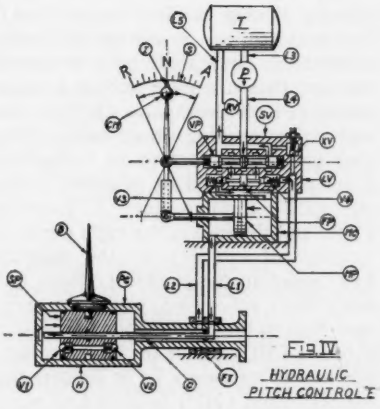


Fig IV
HYDRAULIC PITCH CONTROL

control member (CM) being normally free, this control system is well-adapted for remote control from one or more stations. For operation by remote control, any remote master control member would of course be provided with pitch indicating means, similar to that illustrated.

OPERATION.

The operation of the pitch control system whose components were described above is as follows: All the working parts are assumed to be in mid-position and the blades in neutral pitch, as indicated by their position and the pitch indicator (I) being at neutral position (N) on the scale (S). When the system is in non-operating condition, the selector-valve (SV) is open and fluid passes freely from pump (P) via the selector-valve ports to the return line. The pump therefore normally runs unloaded and absorbs the minimum of driving power.

If the control member (CM) be then displaced clockwise towards the "Ahead" position, indicated at (A) on the scale, the selector-valve plunger (VP) will be displaced to the right, thereby closing the left exhaust port, and fluid will then flow in the hydraulic circuits as indicated by the small arrows. Pressure fluid pushes floating piston (FP) of lock valve (LV) to the right, thus opening check valve (V4) and permitting fluid to be exhausted from the right end of propeller cylinder (PC) through line (L2), valve (V4) and thence via the right ports of selector-valve (SV) to the exhaust line. Simultaneously, fluid pressure unseats check valve (V3) admitting pressure fluid to the head end of the master cylinder (MC). Since the master cylinder is held stationary by its support the master piston (MP) is forced to move to the left, thereby displacing fluid under pressure from the rod end of the cylinder via line (L1) and conduit (C) into the left end of the propeller cylinder (PC). This fluid pushes servo-piston (SP) to the right, turning the blades (B) so as to cause an increase of pitch.

Due, however, to the follow-up movement imparted to the valve plunger (VP) by the master piston (MP), the selector-valve ports tend to re-open as the pitch is changed and unless the movement of the control member is continued, the pitch changing cycle will be terminated by such follow-up movement, due to return of the selector-valve to open-center position, releasing the pressure therein. When this occurs, check valves (V3) and (V4) promptly close, locking the hydraulic fluid in the closed circuits leading to the master and propeller cylinders. Thus, the blades are locked hydraulically in any set position.

To decrease or reverse the pitch, the control member (CM) is simply moved counter clockwise towards the "Reverse" position, indicated by (R) on the scale, whereupon valve plunger (VP) will be displaced to the left and a pitch changing cycle of the same order but opposite in direction will be performed.

Detailed analysis will show that the servo-piston (SP) will follow exactly the movements of the control member (CM). This follows from the fact that fluid is displaced directly from the master cylinder into the propeller cylinder or vice versa. Leakage at the selector-valve will not affect the synchronism. With hydraulic packings now available, which enable many thousands of piston strokes to be performed with substantially no leakage of fluid, the danger of piston leakage is negligible provided the design, construction and installation are correct. However, should any loss of accuracy in the pitch indication occur from leakage due to eventual wear of the packings, etc., synchronism can be quickly restored by reversing the pitch completely. Whenever the servo-piston reaches either end of its stroke, the check valves (V1), (V2) will pass the correct amount of fluid to re-synchronize the system. Checking the synchroni-

zation of the pitch control system would of course be part of the start-up routine, and would be maintained automatically during normal operation. In any event the system can be quickly re-synchronized any time it is in operation. This is an important practical feature which overcomes the drawback in some pitch indicating systems of the prior art in which any loss of synchronism requires considerable remedial measures.

POSSIBILITIES OF PROPELLER STANDARDIZATION.

The controllable propeller system described above is obviously readily adaptable for use with engines having any type of drive, with any size of shaft or propeller, and for replacing fixed-pitch propellers. Hence it provides exceptional versatility and range of application.

A point of considerable practical importance is that this propeller system facilitates standardization of coordinated propulsion systems. The basic design factor of this system is the diameter of the hub, which determines the bore of the propeller cylinder (PC) and the stroke of the servo-piston (SP), and hence the required capacity of the hydraulic apparatus. Considerable variation in the propeller diameter and propeller shaft diameter is permissible for a given hub diameter. For example, a series of five hubs of properly graduated diameters can accommodate propeller diameters ranging from 18 to 54 inches and shaft diameters ranging from 1½ to 4½ inches. The blades can be designed for rotation in either direction. The pitch is of course infinitely variable over the full range required in any case. Hence a few standardized controllable propellers can replace some hundreds of varieties of fixed-pitch propellers in the above range of diameters. It is therefore entirely feasible with this system to standardize controllable propellers and accessory apparatus with obvious economic and practical advantages in their production, application and use.

REDUCTION GEAR TRANSMISSION UNIT.

Fig. V illustrates a special transmission unit particularly designed for reduction drive, and is considered suitable for use as a component part of small engines in place of the reverse and reduction gear units of present practice. The housing of the unit is adapted to be bolted bodily to the flywheel housing in place of the usual reverse and reduction gear unit. It houses the reduction gearing and a hydraulic unit (HU). The reduction gearing is of conventional pinion and bull-gear type and does not require detailed description beyond noting that it is designed for easy accessibility and quick replacement of components, including the pinion and bull-gear, thus permitting ready change of the gear ratio within the usual limits and facilitating repairs.

The hydraulic unit comprises a hollow casing (T) serving as a reservoir for hydraulic fluid and also as a housing for a hydraulic pump (P), of cartridge type, driven directly by the main drive shaft, and the hydraulic fluid transfer components (FT). The latter are mounted around an inboard extension of the bull-gear shaft as shown. The casing is mounted tightly on the partition in the gear housing by cap screws, not shown. Selector-valve (SV), here of known rotary type, is mounted detachably on the top of casing (T). Selector-valve (SV) is pro-

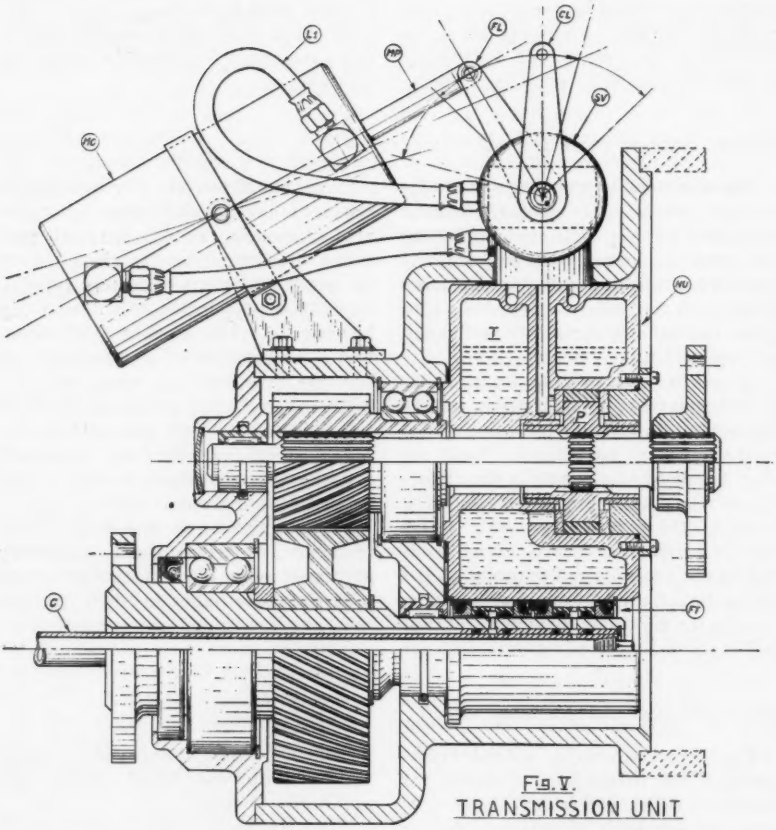


Fig. V.
TRANSMISSION UNIT

vided with a control lever (CL), corresponding to the master control member (MC) in Fig. IV, and a follow-up lever (FL). The master cylinder (MC) may be mounted directly on the gear housing, as indicated in Fig. V, or it may be mounted separately if desired. In either case, the master piston (MP) must be properly connected to actuate the follow-up lever (FL), and the master cylinder (MC) must be hydraulically connected in series with the selector-valve (SV) and the servo-piston (SP) in the propeller as indicated in the schematic diagram. (Fig. IV) The operation of the hydraulic pitch control and pitch indicating system is as before described.

This version would appear to have possibilities of interest in connection with the Navy standard Diesel engines being developed under the post-war engine development program (a).

SPECIAL CONSIDERATIONS FOR LANDING CRAFT.

An important operational requirement for propulsion systems for landing craft is that the engine and propeller should be able to retract the craft after a landing on a beach. *A priori* reasoning might suggest that this requirement rules out controllable-pitch propellers. However, there are several favorable factors to be considered. In general, detachable blades can be made stronger than integrally cast blades. Detachable blades can be made of stainless steel, and can be forged to increase their strength. Furthermore, controllable-pitch blades can be designed for zero initial pitch, that is without helical twist, and the blade section can be made symmetrical, giving the blades great inherent strength due to their form, particularly when turned to low pitch angles. In this connection, it has been proved experimentally that blades designed for zero initial pitch, *when adjusted to optimum operating pitch,* give substantially as much speed ahead as blades designed for the optimum initial pitch, and are superior in astern performance. (b)

The superior maneuverability and greater tow-rope pull obtainable by controllable-pitch propellers are further favorable factors for landing craft. Also, assuming that the propeller can be rotated, the ability to use full engine power even tho the vessel is stopped means that greater power is available to retract the craft. Where the propellers can be used to wash away sand from under the craft, or to wash out a channel to aid retraction, controllable-pitch propellers should provide definite advantages.

It is of course well-known that in powering landing craft, it was found desirable to compound small Diesel engines in twin, tandem and quad arrangements coupled to a single propeller shaft. (a) In such multiple engine arrangements, difficulties are likely to arise in incorporating clutches and reverse gears into the transmission design. It follows that the use of a separate controllable propeller system, as previously described, would probably facilitate the application of multiple arrangements of Diesel engines, and might result in considerable savings in weight, space, and cost of the propulsion system. It is therefore suggested that the use of controllable-pitch propellers with landing craft merits full consideration.

MAINTENANCE, EMERGENCY AND CASUALTY CONSIDERATIONS.

While utmost reliability was the first consideration in designing this controllable propeller system, and in its present stage of perfection hydraulic apparatus provides consistently reliable performance, it may be of interest to examine some of the maintenance, emergency and casualty possibilities.

Calculations indicate that the hydraulic pressure required to operate the pitch changing mechanism, even in the case of large propellers should not exceed 250 psi, hence the hydraulic system can be classed as a low pressure system. This means in practice that the requirements for pumps, cylinders, valves, piping, fittings, packings, etc. are easily met. Since the pitch is locked wherever set, the pitch changing mechanism would be in operation only a small percentage of the operating hours, say 5% as a maximum. With respect to the hydraulic apparatus within the vessel, such apparatus is readily accessible and is also of proved reliability by reason of extensive use

with machine tools and in other exacting applications. Therefore, maintenance and repairs of the hydraulic apparatus should not present any particular problems.

The matter of damage to the propeller blades having been heretofore considered, it only remains to consider the probable results of possible wear or damage to the parts of the hub assembly. The bearing surfaces in this design are ample, and being adequately lubricated, there is no reason to expect abnormal wear The blade mounting sealing ring is well protected against undue wear or injury, and is readily replaceable. Since it is normally subject only to hydro-static pressure of the water above the propeller hub, and the rubbing motion is slow and infrequent, the duty requirements are quite moderate. In this connection, a hydraulic propeller having blade mountings of similar type, but having a blade loading four times as great as for a marine propeller of equal size, was in continuous operation for eleven years before it became necessary to renew the packings and bearings.

The servo-piston packing rings are subject only to low pressure, to infrequent slow motion, and are well lubricated; hence the wear factor is quite small. Moreover, in event of leakage of the piston packings, the blade mounting packings are capable of retaining the hydraulic fluid within the hub, hence there is no danger of sudden leakage in the hub such as would prevent operation of the servo-piston.

In the event the blades strike a heavy obstruction, or the vessel runs aground, damage to the blades or a bent propeller shaft will not impair the operation of the pitch changing mechanism. Should the shock forces be great enough to loosen the joint between the hub and the propeller shaft flange, thereby causing leakage at this joint, the servo-piston can nevertheless be actuated to obtain ahead pitch, since the outer hub chamber will remain intact.

It should be evident from the foregoing description and discussion that there are no adequate grounds for considering that this controllable propeller system should prove any less reliable in practice than other mechanisms now in use for obtaining reverse propeller action. Nor is the apparatus involved more complex and costly than that involved in any other system for obtaining reverse propeller action.

With respect to complete power failure, a stand-by hand-pump, of which various types and sizes are commercially available, can be provided to permit of manual operation in such emergency. This is feasible for even large installations, since when the engine is slowed down the forces to be overcome to change the pitch are of a minor order, and the quantity of fluid to be pumped is relatively small. By way of comparsion, it is much easier to provide for manual operation of a pitch changing system than for a steering system.

ADVANTAGES OF CONTROLLABLE PROPELLERS.

It is suggested that the advantages of controllable propellers can be more readily appreciated and evaluated by a classified list of these advantages such as follows:

Direct Advantages:

1. Increases Thrust/Torque ratio and therefore increases propulsion efficiency under various operating conditions.
2. Eliminates propeller design compromises, and avoids risk of inferior propeller performance in practice due to incorrect pitch.

3. Provides practical means for avoiding critical torsional vibration conditions.

Indirect Advantages:

1. Eliminates need for clutches, reverse gears, direct reversing mechanism, and their accessories.
2. Enables full engine power to be used whenever needed.
3. Improves engine performance, because engine can be run within power, speed, and torque ratings under all operating conditions.
4. Requires fewer engine starts resulting in elimination of engine stalls and reduction of piston ring and cylinder wear.

Over-all Advantages:

1. Assurance or realizing optimum performance of vessel in practice.
2. Less over-all weight, size, and cost of propulsion in many cases.
3. Greater margin of safety in meeting emergencies, superior ease and precision in handling the vessel, fewer and quicker maneuvers, increased acceleration of the vessel, quicker reversing and stopping, shorter stopping reach; superior slow-speed operation.
4. Facilitates direct bridge control of vessel; eliminates engine room delays in maneuvering.
5. Increased dead-pull and towing power; increased speed; increased range; increased operating efficiency.
6. Operating savings with respect to fuel, lube oil, compressed air, engine maintenance and repairs, etc.
7. Possibilities of eventual propeller standardization in certain classes of installations.

In the aggregate, these are substantial advantages even tho all of the advantages enumerated are not realized in any particular case. An excellent and comprehensive report on Navy experiments and tests with controllable-pitch propellers is given in reference (c). While the propellers tested were considerably larger than those here under consideration, the conclusions there stated are considered applicable in the main to controllable propellers for boats and small craft.

A final technical consideration is that marine Diesels, marine transmissions, and fixed-pitch propellers have already been developed to the point where no substantial gain in efficiency or economy in these can be expected. It follows that further substantial gains in operating efficiency can only be realized by complete coordination of marine propulsion systems, based on controllable propellers, as has already been done in aviation propulsion systems.

ECONOMIC CONSIDERATIONS.

It has heretofore been generally assumed that controllable propellers would inevitably involve an increase in the over-all cost of the propulsion system. However, a technical commission appointed to consider the proposed designs for a Swedish government salvage tug and ice-breaker found that substantial over-all cost savings, amounting to 27% of the cost of the propulsion system, and other advantages would be obtained by the adoption of controllable propellers, and they were accordingly specified. A study of costs made by the author and others indicates that substantial cost savings, and reduction in the size and weight of the engine and accessories, can be realized in the case of Diesel installations; mainly due to the elimination of the clutch and reverse gear, and elimination or reduction of accessory apparatus.

Controllable propellers are considered to offer possibilities of con-

siderable cost reduction in mass production. By eliminating hand-work in finishing the blades, the major cost in producing propellers might be substantially reduced. Machine tools for machining single blades are widely used in the production of aviation propellers, and have been applied abroad to the production of controllable marine propellers. Also, since a controllable propeller comprises a separate hub casting and three or four blade castings, the casting process is facilitated, and more precise castings can be obtained. Standardization offers further possibilities of reducing over-all costs.

A further point to be considered is that controllable propellers are functionally useful in improving the operating efficiency and over-all performance during all the operating time; whereas other means for obtaining reverse propeller action are functionally useful but a small percentage of the operating time, and may involve operating losses. When it is considered that the clutch and reverse gear mechanism ordinarily used on small Diesel engines may represent 20-30% of the weight and cost of the engine, yet is useful only about 2% of the time the engine is running, it becomes evident that the clutch and reverse gear is inherently an uneconomic solution of the problem of obtaining reverse propeller thrust.

As compared to electric drive, controllable propellers offer equal or superior results in propulsion, and the advantages of less over-all cost, less weight, space and complexity of propulsion equipment, and greater over-all operating efficiency.

The increased navigational safety obtainable by the wide adoption of controllable propellers would extend potential economic benefits to all classes of vessels, even those not equipped with such propellers. This is a point thought to merit consideration by operators, owners, and underwriters of vessels, as well as by maritime regulating authorities.

CONCLUSION.

In view of its proved and potential advantages, it seems justified to regard the controllable propeller as the most promising development in marine propulsion now in sight. Accordingly, it would seem to merit serious consideration by naval architects, marine engineers, builders of propulsion machinery, builders of vessels, and all others interested in the improvement of American shipping. The extent of the adoption of the controllable propeller, with its possibilities of providing more efficient and economical propulsion of sea-going ships, may well have considerable influence on the future competitive position of the maritime industries of various nations.

REFERENCES.

(a) "Propulsion Diesel Engines for Landing Craft and Small Boats" by Ambrose, Humphreys and Swiderski. Transactions of the Society of Naval Architects and Marine Engineers, 1948 (Paper #4).

(b) "Propeller With Adjustable Blades" by H. F. Nordstrom, Publications of the Swedish State Shipbuilding Experimental Tank, 1945 (Publication #4).

(c) "Controllable-Pitch Propellers", by L. A. Rupp. Transactions of the Society of Naval Architects and Marine Engineers, 1948 (Paper #5).

TESTING ELECTRICAL INSULATING VARNISHES — An exhaustive treatise in which all types of tests of electrical insulating varnishes are discussed as to their *method, variables* and *significance.* In two parts.

BY HERBERT G. STEFFENS.

PART II.

EDITOR'S NOTE: This is the second and final part of the article which was begun on page 375 of the May 1949 issue of the JOURNAL. All references were printed in the Bibliography on pp. 397-406 of that issue.

MECHANICAL TESTS.

Drying Time.

The drying time of a varnish is the time required for an applied film to assume a predetermined degree of structural unity at some definite temperature, humidity, and ventilation rate, at which time there is also developed a general resistance to dispersion or solution by solvents. On the basis of this definition, it is not difficult to appreciate that many different interpretations of drying time are at hand. Each of these depends on what is considered to be some degree of structural unity and what methods are employed in its evaluation.

TEST METHODS. The commonly-used drying time tests are grouped into finger-touch, friction, and adhesion tests, each of which are briefly discussed as follows:

(a) *Finger-touch tests.* These tests are conducted by applying the finger to coated specimens under a variety of pressures. Two conditions of the film are noted, the set-to-touch time, and the dry-hard time. The former is characterized by that point at which a tacky condition is noted but none of the film adheres to the finger. The latter is determined as that point at which a maximum non-rotatory pressure of the thumb and finger does not distort the film or leave marks that cannot be removed by light buffing. Federal Specification TT-P-141a (293) recognizes four distinct conditions of dry as follows:

(i) Dust-free: the finger can be lightly run over the surface without removing the coating; the surface will show no marks, but may be tacky.

(ii) Tack-free: the finger can be lightly impressed without leaving a mark; the surface is not sticky.

(iii) Dry-to-handle: the thumb, applied with pressure in a rotatory manner shows no marks.

(iv) Full-hard: the film can be removed with difficulty using the finger-nail; the film has reached its full hardness.

Regarding the resistance of surface coatings to applied forces, Moore (269) has developed an equation of academic interest, at least, showing the relationship between velocity of flow, viscosity, and applied force:

$$D = \frac{1}{n}(t\text{-}f)$$

where

D = the velocity of flow
t = applied pressure
n = viscosity of the film
f = quantity proportional to the yield value for the film

When the film becomes print-resistant, the quantity f becomes zero, and D is equal to (1/n)t.

(b) *Friction tests.* The resistance to sliding or rolling offered by a coating may be evaluated during the drying period. Although the measurement of sliding friction is not usually made except in certain types of tackiness meters, the rolling friction test, using the Sward Rocker, is commonly met with. This instrument, first calibrated to oscillate a given number of times on plate glass when displaced at a certain angle, is rocked in a like manner on the coated test surface. The number of oscillations to rest is plotted against time, and that point at which the curve so obtained levels off is taken as the drying time. The main objection to the use of this instrument with insulating varnishes is that the number of oscillations is relatively few, and the relative errors are accordingly high. In addition, the continuous motion in the same track registers a false degree of tack in some shorter-oil varnishes.

(c) *Adhesion tests.* The most commonly used drying test is the measurement of resistance to removal of a paper strip applied to the coated film under a constant pressure for a given length of time. Drying is generally considered complete when the paper falls off of its own weight on inversion of the test specimen. The A.S.T.M. test (284, 295) represents the highest development of this test. Here a 1½ inch by 6 inch kraft paper strip is applied under a pressure of a one-pound weight one inch in diameter for one minute. The British Electrical and Allied Industries Research Association (36) and the British Standards Institute (107) use this type of test, except that a No. 4 Whatman filter paper is substituted for kraft paper, and the drying time is taken as that point at which the paper falls freely of its own weight when inverted, without any evidence of adherent lint on the film. Federal Specification TT-P-141a (293) specifies the use of a kraft paper under a pressure of two pounds per square inch.

The Sanderson Test (52), most popular and perhaps the least significant of the purely mechanical tests as far as insulating varnishes are concerned, allows sand to flow on a rotating circular track along the specimen. The length of the track, strictly speaking a spiral toward the center of the panel, is considered the tack-free time. The length of the track remaining after a light brushing is taken as the dry-to-touch-time. Henry (308) describes a dust-free time determination procedure in which an apparatus is used to deposit a determinable and controllable amount of flock to the film.

Stringham's test (28), originally a shop test for drying time, is conducted by bending back upon itself a coated strip and pressing together with the thumb and forefinger. The time at which two test films do not adhere is taken as the drying time. This was also the official Navy test under Specification 52V13 for more than 16 years. Some operators take the endpoint as a separation without audible click, but this point is not very reproducible in the author's opinion. Many tests conducted by the author show that the Stringham test at 20 degrees C. and the A.S.T.M. test at 25 degrees C. are very closely equivalent.

TEST VARIABLES. Among the general test variables may be included temperature, humidity, ventilation, radiant energy, and film thickness. These variables are in addition to such mechanical elements as may be peculiar to the specific method of test employed.

(a) *Temperature.* As noted previously, the process of drying is essentially a chemical reaction of unimolecular order, that is: it follows closely the theoretical mathematical expression for a unimolecular reaction. Therefore, there is a definite relationship between rate of drying and temperature. It is necessary, therefore, that the temperature be closely held to within a degree or so of the established test temperature.

(b) *Humidity.* Very few specifications acknowledge the importance of variable humidity on drying time tests (exceptions, see (293) and 295)). The humidity affects the drying time in two ways: the evaporation of solvent is more rapid the lower the humidity, and the presence of moisture is necessary to reduce the length of time required to initiate the drying period (the so-called induction period). With fairly efficient conditioning systems, there should be no difficulty in controlling the relative humidity to within two percent.

(c) *Ventilation.* Control of ventilation during the drying period is important, as is shown by the data of Fleming and Steel (30) under Figure 3, and by Fleming (133). It is difficult to prescribe a rate of change of atmosphere without also specifying a maximum ratio of film area to drying enclosure capacity. The practice of the author in this matter has been to dry about 40 to 50 square inches of film area per cubic foot of space, using one complete air change per minute.

(d) *Radiant energy.* Tests conducted by Nicholson and Holley (191) show rapid oxidation rates (from gain in weight data) of oil films when subjected to lights of varying intensity. Gardner's work (200) with lights of different wave-lengths indicates that variations in drying time also exist depending upon the type, intensity, and wave-length of the incident light. The author has conducted drying-time tests in ventilated enclosures under low-wattage fluorescent light without any apparent difficulties due to varying results, but it is undoubtedly better practice to conduct these tests in the absence of light.

(e) *Film thickness.* The drying time of varnishes is very sensitive to variations in the film thickness of test specimens. Common practice (284, 295) is to base the test results on a dried-film thickness of 0.9 to 1.0 mil. In addition to the observation of close limits of thickness, it is necessary that the dipping of specimens be done at slow speeds in order that the thickness of film be uniform along the length of the specimen, and that the test will therefore not be dependent on the position of the test weight.

SIGNIFICANCE OF THE TEST. The drying time test, if properly standardized, is a fairly satisfactory index of the quality of a varnish, all other properties being considered. As such it may be used to advantage to compare

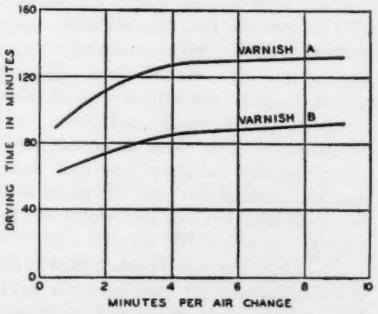


FIGURE 3.—Graph Showing the Effect of Ventilation on the Drying Time of Varnish Films. (After Fleming and Steel)

successive batches of the same formulation for uniformity in production control. For example, the phenomenon of adsorption of driers on the surface of carbon black particles may result in increased drying times during the aging of a pigmented varnish, and the varnish may, as a result, be apparently less resistant to the effects of water immersion. By studying the general resistance of the varnish with relation to its drying time, its performance may be quickly and easily estimated by merely measuring its drying time.

The drying time test data also give an indication of the time required before equipment coated with varnish can be safely handled. This is of particular interest to shop servicemen, to whom efficient utilization of work space is an economic factor for consideration.

Viscosity and Consistency

The nature of the varnish cooking process demands that for a satisfactory definition of a particular material some test must be included which will indicate the relative body or viscosity of the product. That is, some test must be devised which will show to what extent the process of gelation has been carried out in the cooking kettles.

TEST METHODS. Undoubtedly, the most accurate determination of viscosity is that which employs a master viscometer of known dimensions and which operates under the theoretical conditions of true laminar flow. Cannon and Fenske (239) describe an improvement of their modified Ostwald viscometer which permits readings to be obtained on as little as 13 cc. of opaque liquids such as black varnishes with a precision of plus or minus 0.2 percent.

The use of the MacMichael viscometer (284) gives viscosity data in terms of MacMichael degrees, which may be converted into absolute viscosity units using calibrated wire suspensions. The time required for a determination is about one-half hour, and determinations can be made with no consideration of the color of the varnish on about 100 cc.

The Demmler cup (known also as the Westinghouse cup) consists of a brass funnel with a short stem. It is floated stem down on the liquid and allowed to sink under its own weight until a pointed indicator punctures the liquid surface. The time in seconds for the cup to become filled to this point is a measure of the viscosity of the liquid. The Ford cup, another example of short-tube viscometer, is operated by allowing a measured volume of liquid to flow out of a funnel under the influence of gravity. The endpoint is taken as that point at which the exit stream shows a break in continuity.

A convenient method of determining viscosity of transparent liquids is the Gardner-Holdt bubble tubes test. The consistency may be rated as the number of seconds required for a bubble of air of selected volume to rise through the liquid contained in a standard-size tube, or may be compared with a calibrated standard of known viscosity.

A very simple and practical test for viscosity or working consistency is to dip a copper strip in the varnish and allow it to drain and dry in the same position as dipped. The average film thickness is a measure of the consistency, and the difference between the upper and lower film thicknesses is an indication of the amount of drainage or flow during drying. Thus, viscous varnishes will show greater film thicknesses than more fluid ones, and the drainage measurement will indicate the probable distribution of dried film on apparatus when coated in production shops.

TEST VARIABLES. Each method of test described above has variables which are peculiar to it. There are, however, certain general effects which are a function of the property of viscosity. For example, in all viscosity tests, variation in the temperature of the liquid must be reduced to a minimum since the physical nature of viscosity is dependent upon temperature. The viscosity of varnishes generally varies about two percent per degree C at room temperature.

In all viscometers of short-tube construction, maximum accuracy is obtainable only when conditions of laminar flow are realized (164). For this reason, the precision of the Ford and Demmler cups are greatest when the influx and efflux times are greatest. These cups are not recommended for accurate testing except in shop checking, and should not be used to replace the more accurate Ostwald viscometer.

Obviously, using the copper strip, or mil-pickup test, the speed of withdrawal of the strip from the varnish will determine the average film thickness as well as the drainage. To these errors may be added the errors in measurement of thickness. Thus, if a possible absolute error of 0.02 mil is made in each measurement, including that of the bare copper, the aggregate errors may be appreciable. For example,

thickness at panel top = 0.60 mil
thickness at panel end = 1.20 mil

$$\text{Drainage} = \frac{1.20 - 0.60}{0.60} = 100\%$$

$$\text{Possible fractional error} = \frac{8}{60} + \frac{4}{60} = \frac{12}{60} = 20\%$$

SIGNIFICANCE OF THE TEST. Viscosity measurements are perhaps the most widely used tests to compare batches of the same formulation for uniformity in production, and in this respect enjoy a measureable amount of usefulness in varnish testing, particularly where the customer expects a pre-determined quality on shipment.

The consistency test serves as a measure of the degree of coverage to be expected in service application of the varnish. The drainage test shows the tendency towards loss of useful coating solids on draining or oven-baking. These two tests are of considerable importance to the service repairman since low film thickness and excessive drainage usually result indirectly in decreased profits.

The viscosity measurement, in some quantitative form, is the basis of package stability or dip-tank stability tests, particularly where the specific gravity of the varnish is maintained constant or corrected for by addition of solvent (183). Obviously, since the use of a varnish in an emergency after long periods of storage may be occasionally necessary, the stability of a varnish should be such that no significant difference is noted in its behavior to the action of solvents or in its drying or draining characteristics.

Oil Resistance.

In view of the fact that varnished insulation must be frequently exposed to oil splashes from bearings, and is often used to insulate oil-immersed parts of electrical equipment, oil resistance of varnishes is a necessary requirement.

Oil resistance may be defined as the extent of mutual electrical, mechanical, and chemical effects of varnished insulation and oil media on one another during any prescribed period of time at a temperature within the possible operating range of electrical equipment.

TEST METHODS. Numerous methods have been suggested for determining the oil resistance of varnishes, and perhaps no method yet developed has been entirely satisfactory. Some workers, (3, 12, 14) prepare cloth specimens for immersion tests, whereas later methods (28, 55, 60, 284, 295) specify copper sheet as the base material. Flight (10) and the British Engineering Standards Association (84) find more service-simulance in testing coated copper coils, the former immersing the coils for 100 hours at 80 degrees C., and the latter for 24 hours at 105 degrees C. Both specify that the test shall show freedom from roughness and "no appreciable effect on the film". Griffin (60) cites an uncredited source in his method of boiling a mixture of water and oil into which a coated copper strip is placed for two hours.

Many specifications and test methods favor the determination of the degree of oilproofness mainly from film hardness data. Arnold (139) calculates the reduction in hardness due to oil immersion as the percentage oil resistance, although no provision is made for some varnishes which the author has observed to exhibit a consistently greater hardness after immersion. Canadian Specification C-612-A (270) is particularly stringent in its requirement that no softening of the film is permitted.

A possible explanation of the effect of oil on varnishes at elevated temperatures is as follows:

(a) the oil possesses a natural softening, swelling or dissolving effect on the film, perhaps independent of the temperature to some extent

(b) the temperature of the oil bath tends toward development of harder films on continued heating

(c) the superimposition of these two effects is to produce a minimum hardness at or near the 24-hour immersion point.

This explantion, illustrated in Figure 4, holds for a wide variety of varnishes tested by the author.

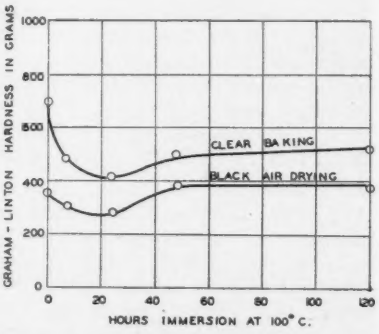


FIGURE 4.—Changes in Film Hardness of Varnishes During Transformer Oil Immersion.

TEST VARIABLES. The variables affecting oil resistance of varnishes are temperature, oil solvency, time of immersion, ventilation of the oil surface, thickness of film, and the age of the varnish film.

(a) *Temperature.* The difference in the solvent action and hardening effect at various temperatures determines the oil resistance of the varnish. It is important that the temperature be maintained at relatively narrow limits. The author uses ovens for this test which hold a temperature of 100 degrees C. within one degree or less. Frequent opening of the oven may be necessary for inspection of the test specimen, but since the mass of oil used is usually great, heat inertia prevents the bath from cooling more than a fraction of a degree.

(b) *Time of immersion.* In view of the data of Fig. 4 it would appear that film hardness varies considerably with immersion time. However, once partial solution or blistering of the film has started it does not disappear. On the

basis of thousands of tests conducted, there has not been a single instance drawn to the author's attention where the film showed signs of failure at the end of 24 hours' immersion where the same effect was not apparent at least to the same extent at the end of a 48-hour period.

(c) *Oil solvency.* The higher the solvency of the oil, the more evident will be any effect on the varnish film. Although no test method is presently known which will determine the solvency of the oil under the conditions of the test, it is reasonable to assume that a solvency test such as the Kauri-butanol test, will indicate serious changes in oil solvency from batch to batch at room temperatures. The fact that the solvency of oil decreases with respect to the time of aging at the normal test temperature is shown graphically under Figure 5. This would preclude the use of one sample of oil for more than one test.

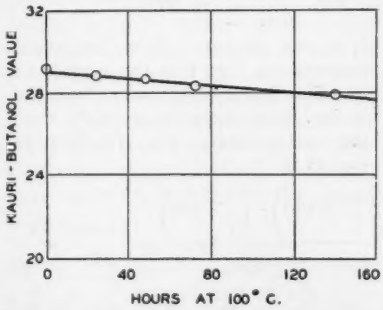


FIGURE 5.—Graph Showing the Decrease in Solvency of Transformer Oil During Prolonged Aging at Elevated Temperature.

(d) *Ventilation.* Resistance of varnish films to the action of freshly distilled and confined oil vapors is a service condition in partially-filled oil transformers operating at elevated temperatures. Therefore, oil resistance tests should be conducted with a portion of the test specimen exposed to confined oil-saturated atmospheres. This test for oil proofness is a delicate one, since many varnishes which show no effects of liquid oil on the film will blister when subjected to oil vapors at the same temperature. Oftentimes failure of the film can be noted at the oil-liquid-vapor interface, whereas the remainder of the film shows no effect of the oil.

(e) *Film thickness.* An exceedingly important variable is the film thickness of the test specimen. Actual tests have shown, for example, that varnishes showing satisfactory proofness based on specimens having an average of 1.8 mils film thickness fail on the basis of films having thicknesses of 2.5 mils. Usually the edges of specimens which have not been trimmed prior to testing will exhibit this effect due to the generally thicker portion of the specimen at the edge.

(f) *Specimen age.* The more completely oxidized the film, the more resistant it will be to the effects of oil. This characteristic of film makes it necessary that specimens be conditioned for a controlled period of time in order that comparative data may be obtained on oil resistance. It is felt that a 10 percent variation in conditioning might not, however, influence the test results to any appreciable extent. As a matter of argument on this point, Findley and his co-workers (24) noted that oil resistance is increased by over-baking the film, and suggest that the test be conducted at uniform degrees of film hardness in relation to surface drying time.

SIGNIFICANCE OF THE TEST. The oil resistance test is a service simulating test. From this standpoint, the failure of varnishes to meet the requirements of the test definitely indicate poor performance in service.

The test is very sensitive to changes in the type as well as the concentration of asphaltum and resin in the varnish, and may be used therefore to determine quality and uniformity of batch production.

Hazeltine (147) believes that the elasticity of the varnish film (not further defined) is a function of the oil proofness. This is generally considered true for black varnishes, since oil resistance and elasticity are poorer as the concentration of asphalt is increased. However, although these two properties may be related empirically, no data is available to substantiate this view at present.

Heat Endurance.

Insulating varnishes should be sufficiently flexible to withstand expansion and contraction of equipment in service, and should maintain this flexibility over periods of extended heat aging. The ability of a varnish to meet this requirement is termed its heat endurance or "life".

Heat endurance is variably defined as the degree of extensibility at break after a predetermined aging period, or the length of such a period before failure is experienced at some predetermined extension. In either case, the life of the insulation is associated with the ability of the film to maintain the breaking point above a certain distention. Schuh and Theurer (127) have shown that most films in service exhibit visual failure when breakdown distensibilities reached a minimum of about three percent.

TEST METHODS. The test methods for determining heat endurance are classified in order of their popularity as mandrel, direct elongation, and thermal shock tests.

Mandrel tests are conducted by bending the film about a rod of specified diameter and noting the presence of cracks in the bent portion of the film. The theory of the mandrel test has been extensively reviewed by Touchin (282), a digest of which follows:

Given a film of thickness, f, coated on a metal plate of thickness, t, which is bent 180 degrees about a rod of diameter, 2r, and assuming that the outside of the plate is extended, but the inside adjacent to the rod is unextended, then

inner length of base $= \pi r$

outer length of base $= \pi (r+t)$

but

$$\pi (r+t) = \pi r + \pi t = \text{extension}$$

and

$$\frac{\pi t}{\pi r} \times 100 = 100 \frac{t}{r} = \text{percent extension}$$

The length of the film after bending will be

$$\pi (r + t + f)$$

and the extension is

$$\pi (r + t + f) - \pi r = \pi (t + f)$$

and the percent extension

$$100 \frac{\pi (t+f)}{\pi r} = 100 \frac{t+f}{r}$$

It is then apparent, from the conditions of the test, that the outside of the film is extended more than the inside, and accordingly more than the metal base. The fraction in excess is

$$\frac{\left(\frac{t+f}{r} \times 100\right) - \left(\frac{t}{r} \times 100\right)}{\frac{t}{r} \times 100} = \frac{t+f-t}{t} = \frac{f}{t}$$

This proves that, regardless of the diameter of the rod, the percent extension of the outside over the inside of the film is a constant for a given thickness of film and metal base sheet

The above development is based on the assumption that the inside of the metal base remains un-elongated, which is of course not necessarily true. If it be considered that a neutral plane exists somewhere in the metal panel, then the actual extensions would be lower than

calculated using the above expressions. It has been pointed out (163) that the elastic limit of the film is exceeded in most bending tests with the result that the neutral plane may shift toward the side of the specimen under compression. Therefore, elongation would be increased, so that the resulting correction may be an undeterminable combination of both tendencies.

The A.S.T.M. (284) and the Navy (295) tests specify a single bend at different points on the test specimen during the aging period. The British Electrical and Allied Industries Research Association (36) creases coated tissue paper back upon itself under a pressure of a one-pound weight. Canadian Specification C-612-A (270) prescribes bending the film 180 degrees about a 3/8-inch mandrel twelve times in alternate directions at the same point on the test specimen, this test apparently intending to measure not only flexibility but fatigue effects as well.

Some workers recognize the main feature of the heat endurance test to be a measurement of direct elongation. Nelson (13) considers the measurement of stress-strain relationships as an important phase of the study of film failures on aging. His data show that stress, y, is related to strain, x, as follows:

$$y = ax^n$$

where a and n are constants depending on the state of oxidation of the film. Thus, on increased oxidation, n increases and a decreases. The experimental data also show that if the varnish film ever approaches perfect elasticity it is only in the early stages of oxidation of the film, and then only at low elongations (299).

The thermal shock test is the most simulative of any service condition met with in the heat aging of varnishes. This test consists in coating specimens on heavy metal bars or rods, and aging at high temperatures. Periodically, specimens are removed and immersed in a cold temperature bath or enclosure. Upon assuming the temperature of the surroundings, the specimens are removed and examined for cracks or flaking. The dielectric strength, using a conducting bath of salt water as one electrode, may be also used to determine incipient flaking or cracking. Niesin (309) cautions, however, that electrical methods usually increase the size of flaws already present and may produce new ones. This test would require for any expression of value a statistical approach involving the measurement of breakdown values of a large number of specimens, due to the widely variable nature of the voltage breakdown mechanism.

Perhaps the main objection to the thermal shock test is that at present insufficient data are at hand by which the results thereof may be correlated with actual life service of the film. Exploratory tests conducted by the author would indicate, however, that varnishes which are relatively poor in flexible heat endurance as shown by the mandrel test are also relatively poor in endurance as shown by the thermal shock test, although the interrelationship between these tests has not been conclusively investigated.

TEST VARIABLES. The factors affecting the test results are classifiable as follows: temperature, film thickness, flexing speed (on mandrel tests), and end-point errors (on mandrel tests).

(a) *Temperature.* Obviously, if the aging characteristics are to be determined at some specified temperature, that temperature should be maintained as closely as possible, say to within two to five percent. The literature is quite controversial on the effect of aging temperatures on flexible life of varnished insulation, since the attempts to correlate data thus far have been in the direction of an overall expression for all varnishes. The temperature effect is a characteristic of the particu-

lar varnish under study and is not necessarily the same for another varnish. Smith and Scott (210) find the following relationship to hold in general:

$$L_2 = L_1 k^n$$

where

L_2, L_1 are life values at temperatures t_2 and t_1 respectively
k is the life constant for the varnish (about 0.44)
$n = (t_2 - t_1)/10$

Rylander's equation (210) is also given as a satisfactory guide:

$$L = K/(t\text{-}a)^2$$

where

K is a constant depending on the construction of the equipment housing the insulation.
t is the temperature in degrees C.
a is a constant for the particular varnish.

Montsinger (210) bases his equation on the tensile determinations of elasticity:

$$L = (7.15 \times 10^4) e^{0.088t}$$

where

t is the degrees C.
e is the natural logarithmic base (2.718...)

The familiar "two-fold increase in rate per ten degree rise" apparently does not hold for aging tests, since the material being tested changes considerably in physical and chemical nature during the test period. Moses, however, (305) feels that this basic rule is adequate for approximating the insulation life on newer types of synthetic insulations.

Temperature variation during bending is also an important variable, since inadvertent warming of the specimen by the hands may result in as much as several hundred hours additional endurance. The only certain way of eliminating this effect is to bend the specimen mechanically in a temperature- and humidity-controlled atmosphere. This equipment is shown under Figure 6.

(b) *Film Thickness.* Undoubtedly, the chief variable in the heat endurance test is film thickness. Thus, in the mandrel test, an expression for the rate of change in elongation with the film thickness may be obtained by simple differentiation of the elongation expression. Thus:

$$e = \pi(t+f)$$

$$\frac{de}{df} = \pi \frac{d}{df}(t+f) = \pi(t+1)$$

Using this expression, the rate of change in elongation of a film based on a 0.005-inch copper foil thickness is

$$\frac{\pi(t+1)}{\pi r} = \frac{t+1}{r} = \frac{1.005}{0.063} = 16.0$$

If the slope of the loss of elongation vs. aging hours curve is small, then the corresponding change in actual life with thickness may be many times this figure shown above. Further argument on this point is based on the findings of Young, Schneider, and Gerhardt (180). Thus, there exists two zones in a film coated on a metal base. The first zone, adjacent to the metal, may be more or

FIGURE 6.—Mechanical Bending Apparatus Used in Mandrel-Tests for Heat Endurance of Varnish Films. (Navy Type)

less oriented due to influence of the metal, and may extend upward to a height dependent on the influence or activity of the metal. The second zone, located directly above the first in the direction away from the metal, is characterized by a manifestation of the true properties of elasticity within the film. With copper metal, the oriented zone is about 0.0004 inch (169). Thus, in a 0.002-inch thick film, only 0.0016 inch contributes to the development of an elastic nature, and any variation of thickness would affect the overall error by a ratio of 2.0 to 1.6 times that calculated by the previous equation.

Changes in thickness of direct tensile specimens are readily detected during measurement of the piece, and accidental variation of thickness is not as important a problem.

(c) *Rapidity of Flexure.* Slow bending in the mandrel test allows opportunity for the film to accommodate itself to the applied stress and permits the film to flow internally. Likewise, slow bending allows loads of greater intensity to be applied than with more rapid bending. Schuh and Theurer (163), for example, recommend 15 seconds to bend films 180 degrees.

(d) *Endpoint Errors.* On the basis of tests conducted by the author, it would appear that the failure points of mandrel tests are not as well defined as in the direct tensile or thermal shock tests. It is the author's practice to coat sufficient test specimens so that one specimen can be bent at three points each day, and the tested specimens discarded. Notwithstanding the fact that these strips may be from several large panels, all coated at the same time under very closely-controlled conditions, unexplainable variations are present in the final test results. The types of breakdown under this procedure are grouped as follows:

Classification	*Example*	*Action taken*
(I) Consecutive	0-0-0-2-2-3-3-, etc.	Fail at last of 2 or first of 3 cracks
(II) Recovered	0-0-2-2-0-0-2-3-3, etc.	Fail at first of 2 or 3 cracks after last recovery
(III) Staggered	0-2-0-2-0-2-3-3-, or 0-3-0-3-0-3-2-3, etc.	Fail at last of 3 cracks of staggered series or first 2 or 3 of consecutive series.
(IV) Unclassified		

The frequency of the group appearance seems to be about 75% in Group I, 20% in Groups II and III taken together, and 5% in Group IV. This last group is composed of cases showing (a) early break followed by extended recovery periods (more than 96 hours), and (b) gradual recovery, such as a 3-2-1- break sequence. Schuh and Theurer, (127) attempt to reduce this spread of failure points by determining the first cracking using a low-power magnification, and are able to determine the true end-point to within two percent. Arnold and Frost (108), on the other hand, find this point either "difficult or impossible" to determine. The author believes that if the test specimens are reasonably selected in random sampling, and are sufficiently similar, particularly with respect to film thickness, then it should be possible to obtain significant data from the mandrel test.

SIGNIFICANCE OF THE TEST. The direct elongation or tensile test attempts to measure with a fairly high degree of precision those tensile properties of a film which contribute most to its heat endurance. Schuh and Theurer (163) are, however, of the opinion that free-film work does not

reveal much information except with materials of low-adherance characteristics. Until the relationships between cohesional forces existant within the film, and adhesional forces manifested between the film and the base are more quantitatively known, tests on free films will not entirely displace tests made on supported films.

The thermal shock test is as nearly a service-simulant test as has yet been devised, although the drop from high to low temperature represents a rapidly-applied stress which has no exact counterpart in service.

The mandrel tests are the most bitterly attacked from the standpoint of service simulance. The argument that insulation does not become flexed in service is, of course, a good one. However, bending of films about a round mandrel is a simple and practical method of elongating the film a predetermined amount in a reproducible manner, and does merit the consideration that a preponderance of varnishes possessing good all-around service records also show relatively long life under the mandrel test.

Hazeltine (34, 132) and Arnold and Frost (94, 108) discredit the mandrel test as not conclusively showing acceptable service heat endurance, and would use the test merely to assist in the laboratory identification of a varnish.

Scott and Thompson (255) describe a heat aging test using small motors. Naturally, this type of test gives much valuable information regarding a varnish, but the approach, is somewhat costly and time-consuming, particularly if failures in operation occur from other causes than insulation failure. In this case, it may be of exceeding interest for a manufacturer to know of actual service usefulness of a particular coil varnish notwithstanding the expense of the test, but it would hardly be convenient for a laboratory testing program involving 20 varnishes per week, for example.

Film Hardness.

Hardness is defined (186) as the resistance which a body of specified dimensions will offer to simple deformation. Its physical evaluation is not simple. None of the various methods which have been suggested for its evaluation are entirely suitable, since few measure the same characteristic of hardness—elasticity, adhesion, abrasion resistance, and penetrability. One must take care, therefore, that a particular type of hardness measurement is specified.

TEST METHODS. The most popular test for determining film hardness consists in scraping the film with the fingernail or with an instrument designed to apply a predetermined pressure in a similar manner. The Graham-Linton Hardness Tester (200) utilizes a blade shaped like a thumb-nail which can be applied to the film under loads up to 2000 grams. Mabb (146) describes an instrument equipped with a needle-shaped pointer which is drawn across the film under an increasing pressure until a scratch mark is left on the film which barely exposes the base.

Abrasion resistance tests are usually conducted by the falling carborundum or mechanical abrader methods. In the former method, carborundum particles averaging about 0.003 inch mean diameter are allowed to fall upon the surface of the film, and the total weight of particles required to just expose the base taken as the abrasion resistance of the coating. In the latter method, instruments such as the Taber Abraser utilize wheels of special composition which abrade rotating specimens, abrasion resistance being determined as the loss in weight of film per 1000 revolutions, or the number of cycles required

to barely expose the base. The latter criterion, while it may seemingly be incapable of close resproducibility, is in effect quite reproducible. The author uses a two-coat system in most abrasion tests, and finds it not only possible to predict the exposure point with a fair degree of accuracy, but can determine the point where the undercoat is exposed through the worn top coat.

Impact resistance tests are conducted by repeating blows at angles to the film using an object of specified size and weight which is dropped to the film from a predetermined height. Schuh (92) describes an instrument which subjects a coated panel to repeated impacts of 150 to 300 blows per minute using a case-hardened steel ball on an adjustable arm wherewith the force of impact can be varied. The author has used a 250-gram steel ball, 1½ inches in diameter and mounted as a pendulum on one end of a steel rod 18 inches long. The blows are delivered from a height of 10 inches and at an angle of 45 degrees to the normal of the impacted surface, the use of a universal joint at the fixed end of the rod permitting free swinging of the ball. Release of the ball is effected using a controlled magnet, since, by this method alone, it was found possible to strike the specimen with a high degree of superimposition.

Rolling friction is determined by the number of oscilations a curved surface will maintain when rocked over the coated surface of the test panel and allowed to come to rest under the influence of friction. Two familiar examples of this test are the Walker-Steel swinging beam and the Sward rocker. Using the former instrument, the beams, which are connected to two small steel balls, are adjusted so that the center of gravity of the apparatus is a few fractions of a millimeter below the center-line connecting the balls. The beam is swung and the time required for the amplitude of the swing to be reduced by 50 percent taken as a measure of film hardness. The Sward rocker, consisting of a double-edge wheel about three inches in diameter and having a low center of gravity, is set in motion on the coated surface and the time required for the amplitude of the swing to decrease a pre-determined amount taken as a measure of film hardness. The instrument is adjusted initially to give hardness readings of 100 when rolled over flat glass panels.

TEST VARIABLES. In addition to the general test variables of film thickness, temperature, and age of the film, each type of measuring apparatus described above has certain pecularities which affect its usefulness.

The knife-edge scraper depends for its usefulness on the speed with which the blade is drawn across the film, and the extent to which the film is to be removed in determining the end-point. It is the author's practice, using the Graham-Linton apparatus, to draw the instrument across the film at about five seconds per inch, gradually increasing the loading until the base material barely shows through. The endpoint should be reproducible to within 20 to 40 grams, depending upon the nature of the film—that is, whether it is brittle, shows poor adhesion, etc.

Use of the standard carborundum test (88) for abrasion resistance requires standardization of the position of the specimen, size of the carborundum particles, and air-pressure, if such is used. Schuh and Kern (88) report an average deviation in 116 tests of about 0.99 percent, whereas Schuh in a later contribution (92) considers the precision as being limited by film thickness measurement to from 2.5 to 10 per cent.

The Taber Abraser (and similar mechanisms) require intermittent attention to the abraded surface, as well as resurfacing of the abrading elements. Also, the particular grinding wheel

used and the pressure with which it is applied to the test surface affect the results obtained to no small degree.

Impact testing features difficulties in closely superimposing repeated blows and in determining the true endpoint, particularly where the test specimen is resilient and shows appreciable cold flow.

Rolling friction tests are fairly precise when conducted on hard films. With the average insulating varnish, however, general softness results in a damping action so great that the possible fractional error is increased to a highly significant degree. Thus, using the Sward rocker, a period of four oscillations may contain a possible absolute error of one swing, in which case the fractional error would be 25 percent. According to the graph under Figure 7, the corresponding error using the Graham-Linton Tester would be 40 grams in 350, or about 9 percent.

Variables common to all types of instruments used are those of film thickness, temperature and film age. Naturally, since varnish films are thermoplastic polymers, temperature changes exert considerable influence on the physical properties. Age of the film manifests itself as increased hardness as the relatively longer molecules of the film are broken down by deteriorating effects of oxidation into shorter and less plastic groups. Thickness variations affect test results in different ways. For example, with scrape tests, it would be expected that film thickness would exert much more influence than with a rolling friction test.

SIGNIFICANCE OF TESTS. The factors evaluated by the thumbnail test are surface hardness, internal hardness, and recoverable surface deformation. In addition, the test reveals qualitative information regarding the adherence of one coat for the other, and both for the base material. Thus if a varnish be coated on both rough and smooth surfaces, hardness tests will generally

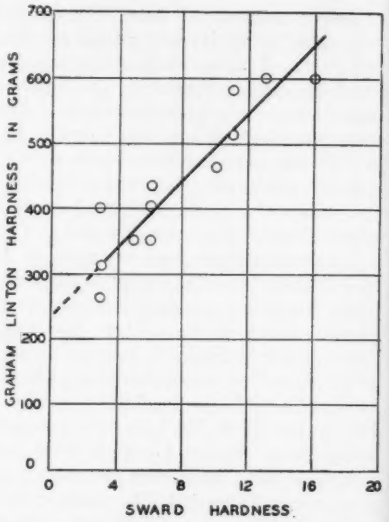


FIGURE 7.—Relationship Between Hardness Values as Obtained Using Graham-Linton and Sward Testers.

show greater values for the rough surface specimen than for the smooth, indicating that the factor of adhesion is taken into account.

The abrasion resistance test measures the work expended in overcoming adhesion of the film for the base. Schuh and Theurer (177) interpret the test as measuring the area beneath the stress-strain curve obtained through measurement of the tensile properties of detached (free) films. Thus, a film of high tensile strength and low elongation may give the same abrasion resistance as a film of low tensile strength and high elongation, if both show the same area under the stress-strain curve.

The impact resistance test fails to give any information regarding the deformation by bodies other than those used in the test. However, it not only indicates initial adherence of the film for the base, but enables the investigator to follow changes in adhesion with time.

Pendulum or rocker tests attempt to

measure all of the following elements: elastic indentation, elastic hysteresis, viscous flow, and frictional and cohesional forces between the contact surfaces (186) The test is, however, based on elements which are not commonly met with in service.

Aside from the measurement of hardness as such, each method discussed above finds some usefulness in following the effects of water and oil immersion. For example, the Graham-Linton tester is recommended (101, 139, 295) for determining the softening point and effects of moisture and oil on the test films.

Water Resistance.

Moisture exerts perhaps the greatest single effect on varnished insulation. Fleming and Steel (30) consider it the principal source of danger to electrical apparatus. Stoppel (40) is of the opinion that if a varnish film absorbs moisture on exposure, the absorbed water will play an important part in its deterioration.

Service reports indicate that the importance of waterproofness is not over-emphasized. Erickson (263) states, for example, that 75 percent of relay and contact failures are open-circuits caused by the action of moisture.

Whitening of the varnish film has been commonly considered the most significant index of water absorption. For example, Stoppel (40) is of the opinion that "in marine work, where waterproofness is the prime requisite, absolute non-whitening may be insisted upon", and Weber (43) considers a varnish film which turns white on water immersion to be of no value for electrical purposes. More recent tests (295) rely on other physical as well as electrical properties to interpret the significance of whitening with the result that opacity of itself is becoming less meaningful.

TEST METHODS. The tests for waterproofness, including resistance to dilute solutions of acids, alkalies, and various salts, group themselves into structural, optical, and electrical tests.

Structural tests are predominantly film hardness tests conducted before and after immersion. Hazeltine (37) describes a test consisting of coating copper ribbons five inches by two inches in size, baking dry, and immersing in boiling water. Film softness is determined by scraping with the thumbnail. The Navy test (295) consists in coating copper sheeting to 2.0 mils in two reversed dips, drying for a prescribed period between coats (24 hours at 25 deg. C. for air-drying grades and 16 hours at 100 deg. C. for baking grades), conditioning for 24 hours at 25 deg. C. and 50% relative humidity, and immersing for 240 hours in distilled water at 25 deg. C., after which period the film is examined for blistering, whitening, and reduction in film hardness.

Tensile tests (299) conducted on detached films show an increase in modulus of elasticity after a period of water immersion, this increase becoming a constant value after about 100 hours immersion.

Tests for whitening on water immersion are seldom of a quantitative nature. A commonly-used test consists in immersing a coated panel in either cold or boiling water for about one-half hour to as long as 72 hours, removal of the specimen and reconditioning for two hours at room temperature. Any whitening remaining after the recovery period is usually reported as permanent effect.

Whitening is due to a condition of immiscibility within the film. Thus, it is believed that the film contains considerable quantities of water-soluble materials formed during the oxidation of the varnish. With the absorption of water, solution of these compounds

take place, resulting in an osmotic pressure developing across the film. Under the influence of this pressure, additional water enters the film, and a condition of immiscibility results.

As shown under Figure 8, light series with the cell is lighted, indicating the passage of a measureable leakage current. Stringham's test (28) is similar, although the use of direct or alternating current voltages is not clearly specified.

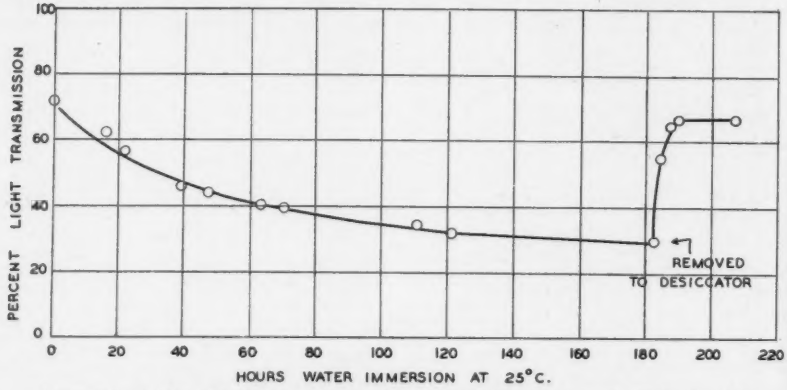


FIGURE 8.—Light Transmission Curve for Determining Degree of Whitening of Varnishes on Water Immersion.

transmission measurements may indicate the degree of whitening together with retained losses in transmission after reconditioning. A more precise test would be perhaps the measurement of light diffusion.

Electrical tests for the lowering effects of water immersion are limited practically to measurements of loss in resistance or dielectric strength. Since the latter have been adequately discussed under a previous heading, no further mention will be made of them here.

Resistance tests are variably conducted on rods, coils, and sheets. One of the first rod tests described is that due to Dean (12). Steel rods, ⅜ inch in diameter and 12 inches long, rounded at the lower end, are dipped in the varnish, applying about 2.5 mils of film in three coats. Specimens are suspended in saturated salt solution, presumably sodium chloride, under 110 volts alternating current. Breakdown is noted as that time at which a lamp in

Partz (55) applies three coats of varnish to polished brass rods, ½ inch in diameter and six inches long, leaving one inch bare opposite the rounded end. Upon immersion in water, 220 volts are impressed across the rods, and failure is noted when three breakdowns occur in 24 hours. Dixmier (70) describes a similar test using battery voltages (2 to 6 volts) to obtain a maximum leakage current of 0.05 ampere at failure. The method described in the Canadian National Defense Specification C-612-A (270) outlines tests on brightly polished copper rods, ¼ inch in diameter and six inches long, coated with two coats of varnish. Thickness of the film is determined after each coat has been baked one hour at 90 degrees C. Before immersion, the rounded end of the rod is dipped in paraffin wax, apparently to avoid occurence of premature breakdowns at the draining end. A voltage gradient of 200 volts per mil is applied for one minute on specimens immersed to a depth of 5¼ inches in

water for 24 hours.

The Navy test (295) consists in coating highly polished brass rods with three coats of varnish, drying each coat for 24 hours at 25 degrees C. (air-drying grades) or 16 hours at 100 degrees C. (baking grades), and immersing specimens for 200 to 300 hours in water at 25 degrees C. A voltage stress of 110 volts direct current is placed across the test specimens, the coated rods being made the cathode of the test cell. Breakdown is indicated by illumination of a lamp placed in series with the cell.

The rod test is also used to measure the resistance of the varnish to water solutions of acids, alkalies, and salts. The placement of direct current voltages across the salt water cell, for example, must be made with an understanding of the electrochemical situation at hand. Thus, a solution of sodium chloride creates a condition of alkalinity about the cathode during operation, and as the specimen becomes a poorer insulator, this alkalinity is increased. Eventually, breakdown occurs as a result of progressive deterioration. It is also interesting to note at this point that recent investigations by Kittelberger and Elm (310) show that the rate and degree of water absorption of varnish films are increased when an electrical potential is applied across them.

Measurement of insulation resistance after water immersion, using guarded electrodes applied to the specimen, is undoubtedly a more precise determination of water resistance. The Navy test (295) utilizes a thin, coated copper sheet about 3¼ inches by 4 inches, the film coated on one side of the copper only. Guarded circular electrodes are placed on the film using a thin application of petrolatum as an adhesive, the copper sheet acting as the other electrode. Insulation resistance is measured using a direct current voltage of 180 volts. As a rule, voltages in excess of 200 are to be avoided, since the wet dielectric strength of many varnishes is relatively low (see Figure 13).

Although the above tests account for most of the methods now used to determine water resistance of varnishes, some special tests of more or less service simulance have been described. Thus, Weightman (14) impregnates coils composed of 10 turns of double cotton-covered wire and places them across a 100 volt source, using as a second electrode a dilute solution of salt in which the coils are placed momentarily. The precision of such tests is, however, low, since there is no way in which the film thickness of the test specimen can be controlled or measured, or in which the leakage current can be prevented from influencing the test results in the absence of proper guarding.

TEST VARIABLES. Various factors affect the reproducibility of the water resistance test, among them baking temperature, film thickness, purity of water, and composition and cleanliness of the base metal on which the specimen is prepared.

Variations in temperature affect the rate of imbibition of water by the film. Thus, where the immersion period is definitely prescribed, relatively high temperatures may result in more rapid approach to saturation conditions within the film. Temperature stabilization in the measurement of tensile and electrical properties is, of course, necessary. Generally, the temperature should be held to within 10 percent or less, and preferably to within two degrees or less. C.

In cases where a prescribed drying period is given, the completeness of oxidation of the film is closely associated with film thickness, and in this respect, controlled thickness of film is very important. According to Kendall (272), the weight of a film (equivalent to film thickness in films of similar specific gravity) and water absorption vary as shown by the following empirical expression:

$$\log W/V = \log W_0 - (1\text{-}n) \log V$$

where

W = water absorbed per unit area
V = film weight per unit area
n = constant for a particular varnish
$\log W_0$ = log W when log V is zero

The film thickness of rod specimens prepared as noted above varies along the rod, since dipping, draining, and drying are effected in the same direction. Thus, tests conducted show the average thickness at various points along the rod to be as follows for the three individual coats:

Inches along	*Film thickness in mils*		
	Coat 1	*Coat 2*	*Coat 3*
0	1.00	1.32	2.87
1	1.05	2.02	3.30
2	1.00	2.15	3.32
3	1.00	2.15	3.45
4	1.00	2.42	3.60

This tendency toward non-uniformity in film thickness on rod specimens contributes heavily toward variability of the test. Attempts to reduce this variation by inverting the test specimen between alternate coatings, or during the draining period have been no apparent help. However, as the dipping speed approach the draining rate of the varnish, the tendency toward wedge-shaped films becomes less.

Only distilled water should be used in water immersion tests. The presence of salts in the water tends toward reduction in the rate of imbibition by the film. This effort is substantiated by the results of whitening tests conducted using both distilled and saline water. Films known to whiten rapidly in distilled water showed no such effects in three percent salt water over a given period of time.

As noted previously, solutions of salts develop effective alkalinities at the cathode of resistance test cells, and therefore, the Navy sea-water proofness test is essentially an alkali test. Any change in the components of the solution must exert some corresponding change in the effective alkalinity. Thus very hard tap water would probably reduce the apparent resistance of rod specimens immersed therein.

Extreme pains must be taken in the preparation of water resistance test specimens. Variable roughness of metal surfaces (12, 28, 55, 70, 270, 295) influences the resistance to a marked degree. The author has noted repeatedly that varnishes which would normally pass the Navy sea water proofness test using etched rods correspondingly fail when applied to highly polished rods.

Traces of salt residue from fingers on rods also cause trouble, since varnishes showing inherent blistering tendencies almost always develop blisters first on soiled areas. Apparently, the salt deposit increases the osmotic pressure developing across the film at that point, and greater amounts of water are permitted to enter through the film.

SIGNIFICANCE OF THE TEST. Physical tests for the effects of water immersion make it possible to evaluate the probable resistance of wet films to mechanical shocks and stresses. Thus, cleaning of electrical apparatus after periods of water immersion can be safely accomplished if the abrasion resistance of the film is greater than the stresses developed during the operation.

Examination for blistering is of especial significance, since the presence of blisters, aside from being good secondary evidence of the deleterious effects of water immersion, points toward a potential source of danger. Thus, even slight abrasion will probably be sufficient to puncture the film, causing insulation failure. For this reason, blistering in any form is the subject of concern in varnish testing.

The significance of whitening is treated controversially in the literature. Older views on whitening seem to be

displaced in favor of later studies. The measurement of post-immersion electrical properties is now being relied upon almost entirely to indicate reliable performance of the varnish. Gardner (91) considers the whitening of a film as merely a change in refractive index of the film, and that it is "in no way a measure of its utility as a protective coating".

Analysis of loss-in-weight data on water immersion is quite difficult to interpret unless some information is at hand concerning the chemical composition of the varnish, in which case then it may be good corroborative evidence. Thus, Rinse, Wiebols, and Takes (184) find leaching losses as high as 29 per cent for limed rosin varnishes, whereas phenolic varnishes show as little as 5 per cent. Water absorbtion data on these same varnishes show 344 per cent and 2 per cent, respectively. This substantiates the common opinion that phenolic resin varnishes are generally superior for water resistance, and that limed rosin varnishes are inferior.

Reconditioning of blistered immersion specimens often results in complete disappearance of the blistered areas within four days at 25 degrees C. and 50% relative humidity. This action may be interpreted in two ways. It may supply information regarding the ability of the film the recover from the immersion effects, that is, its ability to maintain flexibility even after prolonged periods of immersion, or it shows that the film is relatively permeable to moisture even after normal conditioning, since the speed at which the blisters disappear is roughly proportional to the water-vapor permeability of the film. Obviously, the former interpretation places emphasis on the utility of the film as a stable material, whereas the latter, for the same action, illustrates the undesirability of the film from the standpoint of vapor porosity. This example of diametrically opposite choice of interpretations illustrates the inadvisability of judging performance characteristics based on the physical effects of water immersion alone.

Water Vapor Permeability.

The study of the effects of water vapor on insulating varnish is important, since operation and storage of electrical equipment in relatively high-humidity atmospheres is a common service condition.

The water vapor permeability of a material may be defined as a rate of transfer of moisture in the vapor state through a prepared film of known dimensions under a constant and measureable pressure differential.

Payne and Gardner (171) have advanced a theory of fibrillary structure which is intended to explain the mechanism of permeability. Porous films are caused by separation of fibrils of which the film is assumed to be composed. Under the influence of moisture and associated pressure differentials, there ensues a filling of these pores with water. The process of filling and emptying depends on adsorptive forces between the material and the liquid. This theory is further developed by Carson (179) in an attempt to explain why simple diffusion alone is not sufficient to account for the relatively high permeabilities of some substances. Thus, it is believed that the regions between crystallites (fibrils) provide room for migration of the liquid molecules acting under the influence of adsorption forces. Later thought (280) indicates that the process might be one of Poiseuellian flow through capillary holes in combination with activated diffusion, in which holes formed as the result of molecular vibration fill with liquid and shift to

the side of lower humidity, releasing moisture to the atmosphere of lower vapor tension.

TEST METHODS. The determination of water vapor permeability requires the preparation of a suitable film of uniform thickness and homogeneous structure. One method consists in coating some base from which the films may be readily detached. For example, Bayer and Kampf (172) describe a method of coating aluminum with varnish and removing the base with dilute hydrochloric acid containing platinic chloride. According to New (166), films are not affected by preparation over mercury or amalgamated plates. However, differences of opinion presently exist regarding the equivalency of results of tests conducted on detached and coated-base films. Payne and Gardner (171), for example, show that some very significant differences exist between specimens prepared in each of these manners. The author describes the preparation of detached films over mercury, but under conditions of film tension (299). Payne (205) describes permeability tests conducted with respect to base materials composed of vegetable parchment and glass cloth. The cup method enjoys a measureable amount of popularity in water vapor permeability tests because of the ease with which measurements may be made and the minimum amount of calculation required for the proper interpretation of test results. The Payne method (165) makes use of a small flanged cup weighing about 70 grams and having an inside diameter of about 1½ inches, over which is placed a sheet of the test material held fast to the flanged ring by a fitted gasket. The procedure is to place about five cc. of water in the cup, seal the film securely over the opening, and place the cell in an atmosphere of controlled low humidity. Periodic weighings of the cup give the amount of water vapor permeating the film in any given time.

Some cup methods, such as the Thwing-Albert, specify the use of dessicant in the cup and in contact with the film, the cup being then placed in a high-humidity atmosphere.

Doty, Aiken, and Mark (280) describe a manometric method of measurement which determines the pressure differential developed across a film when one side is maintained at pressures of about 10^{-4} millimeters mercury and the other at the pressure due to the vapor of the liquid at the test temperature. The rate at which the vacuum is degenerated by the permeating vapor is calculated from pressure data. Further computations enable the operator to arrive at an expression for the permeability constant, as well as a diffusion constant and a solubility constant for the test material. Films are supported on wire gauzes and need not be exceptionally strong in that relatively little pressure differentials are ever developed across the film.

TEST VARIABLES. Cup tests are influenced by a number of variables, some of which are instrumental in nature, and some environmental. They include the relative humidity of the enclosure, area of the specimen exposed, ratio of area of liquid to area of specimen, height of specimen above the liquid in the test cell, the establishment of constant transfer conditions, edge-leakage, mechanical leakage of moisture along the flange interfaces, and atmospheric conditions of temperature and relative humidity in the test room. The relative importance of each depends primarily on the particular design of the test cell.

The relative humidity within the cell is closely related to the distance between the specimen and the liquid level within the cup. Some workers prefer to reduce this distance as much as possible, thus reducing the effect.

The area of the test specimen exposed is related to the area of the evaporating surface, since conditions

of saturation are best maintained when a large momentary supply of vapor is at hand against a small vapor leakage through the film.

The time required to arrive at equilibrium moisture conditions within the film itself is a function of the saturation rate of the film itself, and this is in turn dependent upon the relative humidity and the temperature. Most workers ignore this induction period, and base their calculations of permeability on the rate of loss in weight at the 48-hour period.

Edge leakage through the specimen may or may not be a significant factor, depending on the ratio of length of travel through the edge to the length through the film. Thus, in most cases, the ratio of flange length to film thickness is about 100 to one, and therefore, the edge leakage may amount to only about one percent of the total loss, making this error negligable in the light of general discussion.

Mechanical leakage is caused by discontinuous contact of the film with the metal flange. Various methods are practiced to reduce this leakage, among the more common being the use of wax or petrolatum as an adhesive between the specimen and the flange.

Regulation of atmospheric conditions will, according to vapor pressure laws, regulate the pressure differential maintained across the test film. Measurement of actual pressure difference is not possible with cup methods, since the difference between actual and calculated vapor pressures in the cup is not known as a function of the rate of permeability through the film under the conditions of test. Only in cases where the loss in pressure due to permeation is more than compensated for by free evaporation from the liquid surface can the assumption of constant or determinable pressure difference across the test film be made with any degree of accuracy.

Relative efficiencies of desiccants is also a factor in controlling humidity within the test cell, since some desiccants exert nearly theoretical control over atmospheric vapor even after absorbing many times their weight of moisture.

The precision of manometric methods rests solely on the fineness of control exercised over atmospheric conditions, as well as general manipulative technique in the measurement of pressure. The precision of this method is considerably above that possible with the cup method, and, therefore, the permeability values obtained are more significant, although the cost of apparatus used in the determination is high.

SIGNIFICANCE OF THE TEST. The water vapor permeability test attempts to evaluate a characteristic of the material which can be used to compare relative moisture resistance properties of similar materials. Obviously, varnishes of relatively high permeability are undesirable as insulators, not particularly because of the usual deterioration of the film in the presence of moisture, but because of failures which are so often attributed to entrapped moisture in the insulated part. Thus a film may readily permit water vapor to enter an element, but as the film ages and becomes less readily permeated, the moisture is not as easily removed, and the result is that the entrapped moisture facilitates corrosion of the part.

Permeability measurements may have some application in determining heat endurance of varnish films, since microscopic cracks and fissures due to loss in elasticity of the film may be readily detected in the rate of loss in weight as plotted against aging time.

Stability and Corrosive Effects.

Stability and corrosive effects are related properties of varnishes, the latter being dependent in some manner on the former. Thus, if a varnish is chemically stable, decomposition into acids and aldehydes of low molecular weight is slight, and the corrosive potentials are low. However, since many of the tests for stability are either measurements of acidity or of the corrosive effects of acidity, this discussion will center mainly about corrosive effects, rather than stability itself.

Early attempts to define a satisfactory insulating varnish (3, 6) specifically required a non-corrosive effect on copper. This was undoubtedly a reasonable demand, since many of the early natural gum insulating varnishes were of such high acidity that corrosive action on small diameter magnet wire was most pronounced. Warren (98) later concluded that greening action of varnish did not necessarily denote weakness, since the green compounds formed were usually non-conducting. The experiments of Spielman (103) indicated that the greening action stopped as soon as moisture was removed from the film, and that in this respect moisture could be linked directly with corrosion. Rossi (158) conclusively showed that there was no certain relationship between acidity of a film and its effects on metals, although pH measurements would appear to be fairly accurate indices of corrosive nature, as shown by his data in Fig. 9. Stephens and Gehrenbeck (225), on the other hand, discredit the relationship between pH and electrolytic corrosion, stating that it is immaterial whether the anion formed comes from an acid or a salt.

TEST METHODS. Corrosive effects may be measured by coating electrodes and maintaining a certain direct current potential difference between them in the presence of moisture. Weightman (14) uses taped copper strips impregnated and placed in juxtaposition for 14 days, after which period insulation resistance measurements are made between the strips. Visual signs of corrosive activity are obtained by opening the taped strips. Stephens and Gehrenbeck (225) describe a test using two bare parallel wires (#40 or #38) at a potential difference of 90 volts direct current held in tension on the surface coating to be tested. Conditions of 120 deg. F. and 90% relative humidity are maintained for five days, after which the extent of corrosive action is determined miscroscopically.

The Navy test (256) is similar in principle to the above test, except that the extent of corrosive action is determined by measuring the change in resistance of the wire using a Kelvin bridge.

One of the methods described by Thompson and Mathes (297) consists in exposing a sheet of varnish to a combination of controlled temperature and a humidity of 100 percent plus dew for 24 hours. The test specimen is then fitted into a guarded electrode

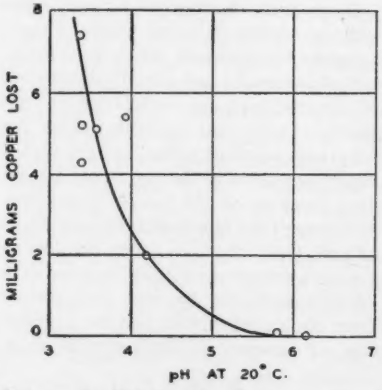


FIGURE 9.—Relationship Between pH of Varnish and the Corrosive Effect on Copper. (After Rossi)

assembly and the current resulting from application of 120 volts direct current difference across the specimen is measured, using a sensitive galvanometer. The characteristic decrease in current after voltage is applied, together with the recovery characteristic after voltage difference has been removed are studied as a function of the corrosive action. Materials which are relatively corrosive show large decreases in current as well as the largest initial current values.

Several methods are suggested for measurement of acidity of the varnish film as an indication of its corrosive activity. The British Standards Institute (107) specifies the digestion of detached films in 25 cc. of distilled water at 70 deg. C., for one hour, after which the pH of the water is determined. Thompson and Mathes (297) stir the film in water at room temperature and determine the conductivity at three volts and 60 cycles.

The U. S. Army Ordnance Department has experimented with a distillation test consisting of placing a weighed amount of varnish in an Erlenmeyer flask and curing for a prescribed period of time. The flask is then fitted with a reflux condenser about the lower end of which is wrapped a weighed metal foil. The flask is then placed in a constant temperature oven at 135 deg. C. and the contents allowed to reflux. The metal foil is weighed after a period of refluxing, and the change in weight interpreted as corrosion of the metal. Exploratory tests conducted by the author indicate that check tests using this method can be obtained with a standard deviation of about 8 per cent.

Distillation tests of a different nature have been conducted by the author on various grades of varnish in order to determine the relative proportion of each volatilized at 135 degrees C. Weighed panels are placed in a controlled oven and aged for 150 hours. Periodically during the aging, specimens are removed and weighed, a graph being plotted to show loss in weight as a function of time. The results of these tests conducted on a heat-reactive baking varnish and an air-drying varnish are shown under Fig. 10.

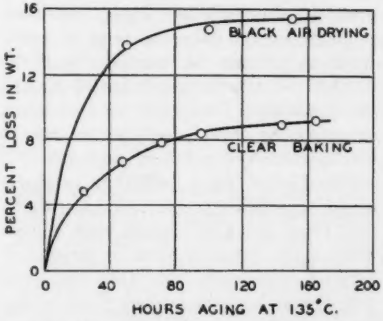


FIGURE 10.—Curves Showing Volatilization Effects of Heat Aging on Varnish Films.

In connection with stability tests, the majority of test methods are viscometric in nature. A sample of varnish is checked for specific gravity or non-volatile content, and the viscosity measured. The sample is then stored in a sealed container and after a predetermined storage period the sample is readjusted to the original non-volatile content and the viscosity again measured. The increase in viscosity caused by spontaneous polymerization is a measure of the package stability of the liquid varnish. By conducting the same test in open containers an indication of open dip-tank stability is obtained A method described by Cotten (183) consists in aerating a sample of varnish at 40 deg. C. to accelerate the test.

TEST VARIABLES. The main source of error in resistance tests on corrosive action lies in the selection of too large a conductor so that the corroded part does not affect the resistance of the wire within the limits of precision of the measuring method. Thus, Spielman (103) observes that visible greening may be the result of as little as 0.001

inch of corrosion, which may not be effective in changing the resistance of any but the finest wire.

Distillation tests, wherein the decomposition products are continually refluxed over various metals, are affected by the position of the metal with respect to the refluxed liquid, the temperature of the bath, the ratio of metal area to volume of varnish, and the method of attaching the metal foil to the apparatus. Care must also be taken in removing metal foils so as not to lose particles of reaction products.

Distillation tests, wherein products of decomposition are completely volatilized from the film require that a constant ratio between area to weight of film be maintained and that the oven ventilation be adequate to remove the decomposition products as they are formed.

The chief source of error in stability tests is the loss of solvent due to evaporation. If hermetically sealed containers are not available, the loss must be made up with solvent which compares in specific gravity to the original thinner. This requires that some volatile must be recovered from a separate portion of the varnish and used to make gravity adjustments during the test. A better method might be to study the relationship between viscosity and non-volatile content, and to adjust the varnish to its approximate original consistency at each test, correcting for differences in non-volatile content according to the calibration curve.

SIGNIFICANCE OF THE TEST. Measurement of electrolytic corrosion of varnished insulation is a service test in every respect. It affords some indication of what part entrapped moisture will play in the eventual breakdown of the insulation during service.

In connection with acidity determinations, the measurement of pH merely indicates that the hydrogen-ion concentration is of a certain value when the specimen is leached in water. Both Spielman (103) and Rossi (158) agree that no certain relation exists between corrosive action and acidity or acid number, which is in contrast to earlier views (43, 50) that greening should be interpreted as evidence of corrosion. Corrosion, in its strictest sense, implies continued erosion, whereas the action of varnish decomposition products on copper for example, is one of formation of a protective coating of relatively good insulating characteristics.

Distillation tests are very important, particularly on varnishes to be used in impregnating parts of poorly-ventilated units. The volatilized products are deposited on relatively cooler parts of the equipment, such as bearings and contact points, and may be responsible for serious damage. It is quite difficult to interpret the results of distillation tests using the change-in-weight data of the reflux test, since many compounds show gain in weight during the test. If any particles should be inadvertently lost during the test, the resultant weight is in error by this amount. Also, if the metal salt of the particular acid evolved is of low molecular weight, the data may be misleading for comparison between varnishes. It is the author's opinion that the distillation test involving refluxing is at best a qualitative one, and will probably not come into more common use until some of the above noted objections are removed.

Distillation tests, wherein the products of decomposition are removed by heat volatilization, indicate rather conclusively the heat stability of the varnish. Also, since almost without exception, the decomposition products evolved at 135 degrees C. are low molecular weight acids and aldehydes, and since these materials are highly polar organic compounds, comparison of the amounts volatilized permits con-

clusions regarding latent corrosive properties of the film.

Package or tip-tank stability of insulating varnish is an important characteristic to the consumer. It indicates how long a varnish can be expected to maintain its fluidity on exposure and to what extent application practice will have to be modified to compensate for progressive polymerization. The gradual increase in viscosity at constant non-volatile content is a measure of room temperature polymerization rate, and has certain definite bearing to the actual cost of the varnish and perhaps profits realized from its use. Hazeltine (34) believes that 30 days stability to exposure in an open dip-tank without curdling or livering is a reasonable expectation, if evaporative losses are periodically compensated for.

Contamination.

Contamination may have an effect on service performance of insulating varnish, depending on what specific type the contaminant is. As a matter of fact, most large repair shops using insulating varnish have filtering facilities for removal of foreign materials from dip-tank varnishes.

From a testing standpoint, contamination is highly objectionable since it prevents attainment of a high degree of accuracy. Thus, contamination makes it difficult to adjust the varnish to the proper test consistency, displaces dielectric test electrodes an indeterminable amount from the film, causes premature breakdowns on rod specimens for water immersion tests, and accounts for questionable failures in heat endurance tests.

TEST METHODS. The author's test for contamination consists in coating a clean glass plate 3½ inches by 8 inches by ⅛ inch with one coat of varnish as received and also at the test consistency obtained by thinning. The number of particles per square inch is determined by simple counting. The purpose of using counts at two consistencies is to obtain the particle count in the original state, and also to determine the effects of thinning to produce seeding out of improperly fluxed gums and asphalts.

TEST VARIABLES. The presence of dust in the drying air, of course, results in high counts. However, if a similar-sized specimen is dipped in a mixture of naphtha and oil, for example, and allowed to remain alongside of the test panel, a control count may be made for purposes of correction.

SIGNIFICANCE OF THE TEST. Contamination counts are reliable indices of the quality of production in that the greater the pains taken to clarify a varnish before shipment, the lower the count. Also, variable counts from shipment to shipment may indicate that the product is not subject to rigid clarification or adequate inspection at the point of manufacture. Contamination counts also assist in explaining certain erratic test results such as premature heat endurance failures or low wet dielectric strength test precision.

Hiding Power.

Black varnishes are generally used in preference to clear types to cover unsightliness in coils and, therefore, a hiding power test is usually necessary. In view of the fact that most black varnishes are solutions of pitch, the element of coverage is not considered in the same respect as when testing pigmented products. Hiding power becomes, in effect, low light transmission in the visible portion of the spectrum.

TEST METHODS. Checkerboard methods for determining hiding power are not very reliable for the purpose intended here, since it is almost impossible to reduce errors of vision to the point of insignificancy. The author has, therefore, measured the percentage light transmission through a 0.002 inch film coated on glass using a photoelectric cell, an incandescent light source of about 135 foot-candles intensity, and a sensitive galvanometer. By correlating light transmission values with a checkerboard reading, it was found that complete hiding occurred at transmission values of from 0.5 to 0.8 per cent. The bulk of black varnishes tested showed transmission values much lower than this.

TEST VARIABLES. Variations in film thickness can produce highly magnified changes in light transmission, depending on the thickness. This is in accordance with Lambert's law, which shows the transmittancy to be an exponential function of the thickness.

Light transmission is also affected by changes in the spectral composition of the measuring light, since components of the varnish, particularly the pitch, absorb more light of higher frequency than of lower. A spectrophotometric transmission curve for a black air-drying varnish containing pitch is shown under Fig. 11.

SIGNIFICANCE OF THE TEST. Aside from measurement of suitability of a varnish for coil coverage, light transmission tests, particularly of the spectral type, may indicate gross changes in pitch type and concentration. In this respect the test may be useful in checking uniformity of quality.

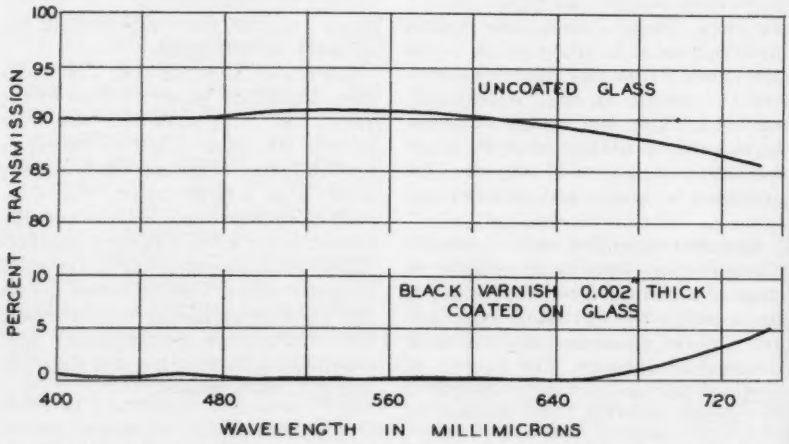


FIGURE 11.—Spectral Transmission as a Means of Determining Hiding Power of Black Varnish Films.

CHEMICAL TESTS.

General Tests for Identity.

Chemical tests on insulating varnishes are of little value except that they may give some indication of the qualitative composition of the material. It is possible, in most cases, to determine the nature of the resins, thinners, and driers present in the varnish, although the formula cannot be derived from this data.

Considerable support can be found for the opinion that the complete quantitative (and in some cases even qualitative) analysis of a varnish is not possible (4, 22, 41, 54, 59, 60, 72, 93, 128, 211). It is, however, still believed by some (306) that chemical analyses should form some part of a specification or quality testing plan.

The author has conducted the following tests on varnishes mainly for the information they give regarding the nature of the material:

(a) saponification number
(b) unsaponifiable content
(c) acid number
(d) non-volatile content
(e) resin tests (Gibbs and Millon tests for phenolic resins, Lieberman-Storch test for rosin and rosin esters, phenol test for phthalates)
(f) ash content and composition (test for driers)
(g) methanol and benzol tests on recovered volatiles.

On the basis of these tests the operator should be able to determine approximately the nature of the varnish, that is, whether it be a phenolic or rosin type and approximately what the oil length is. This latter information may be obtained from the saponification number and the unsaponifiable content. The amount of driers is often a good indication of the nature of the oil, since soft oils such as linseed and soya require relatively high concentrations of drier to provide adequate drying.

THINNERS.

Satisfactory performance of insulating varnishes depends to a great extent on the solvent used. This is particularly true in cases where the varnish tends to case-harden during drying and retains solvent in the film interior. It is interesting to note in this respect that solvents have been found to remain in coil interiors for months after impregnation (226). The greatest danger associated with retention of solvent is the possibility of softening or dissolving magnet wire enamels.

Considerable emphasis is placed on the general problem of magnet wire attack. Peaslee (21) in 1922 was one of the first to caution against selection of compounds or impregnation methods which would cause injury to enamel coatings. He found petroleum naphtha the least injurious thinner. Hazeltine (34) concludes that turpentine is just as dangerous in this respect as high-solvency coal tar thinners. Halls (234) and Underhill (249) classify all coal tar thinners as detrimental and unsatisfactory from the standpoint of magnet wire attack.

The problem of magnet wire attack is made especially critical since the enamel used to insulate bare wire may be of all degrees of resistance to softening by solvents. Arnold and Frost (94, 100) warn that large percentages of magnet wire offered on the market are seriously injured by treatment necessary to impregnate coils.

Certain types of thinners are also responsible for lifting of undercoats in multiple-coat applications. Hofmann and Reid (75) found petroleum hydrocarbon distillates to be the least offensive from a lifting standpoint, and coal tar (aromatic) hydrocarbons to be very much less desirable.

Addition of water-miscible solvents such as the simpler alcohols, increases the possibility that the dried film will have decreased water resistance.

TEST METHODS. Tests are best conducted on portions of volatile recovered from the varnish by steam distillation. This method avoids the possibility of including heat-decomposable substances as might be obtained in ordinary distillation.

Tests to determine the effects of thinners on magnet wire include those designed to measure solvency. The most popular solvency test is the kauri-butanol test, which measures the amount of thinner tolerated by a 16⅔ percent solution of selected kauri gum in normal butanol. The endpoint is taken as that point where 10-point type is barely distinguishable through the body of the titrated liquid (136). Baldeschweiler, Morgan, and Troeller (167) find a more precise endpoint obtainable if titration is continued to the start of the first permanent turbidity.

Arnold and Frost (108) describe a practical test for varnish solvency consisting of immersion of an enamelled magnet wire specimen in varnish for 16 hours at 80 deg. C., after which period the enamel is tested for softening effects by wiping with a soft rag.

The author has conducted tests on various thinners by immersing samples of enamelled magnet wire in the solvent for 168 hours at 25 degrees C. After immersion, and while still wet, abrasion resistance of the enamel is measured using an instrument applying an increasing load to the enamel by means of a constantly-renewed steel music wire stretched perpendicularly across the enamelled wire. The graph under Figure 12 indicates conclusively that the reduction in abrasion resistance of immersed enamel is a function of the thinner solvency.

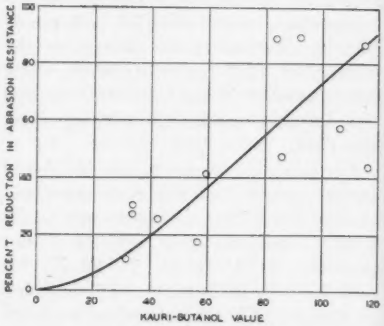


FIGURE 12.—Graph Showing Effect of Solvents of Varying Kauri-Butanol Value on the Abrasion Resistance of Magnet Wire Enamels Immersed Therein.

The author has also conducted tests for lifting action of various thinners on varnish specimens and has found that lifting resistance, as measured in time to produce lifting, is not proportional to the measured solvency of the thinner. The tests were conducted using a small flat copper screen, about ⅞ inch in diameter and connected to an optical lever so that any swelling of the film could be followed and determined accurately. The screen was placed on the film and about 0.5 cc of thinner placed on the screen, the thinner being thus held against the film by capillary action.

A common test for detecting and estimating the presence of aromatic thinners is the sulfonation residue test. This consists in vigorously shaking 25 cc. of thinner with 70 cc. of concentrated (295) or fuming (293) sulfuric acid and allowing the unpolymerized material to rise to the surface of the reaction flask. The volume of sulfonated residue can then be read directly from

the apparatus and expressed on a percentage basis. This residue represents the amount of saturated hydrocarbon compounds which do not enter the sulfonation reaction.

Since turpentine is easily sulfonated in the manner described above, the use of a confirmatory test for aromatics is required. The butanone test used in this case, is in effect a nitration of the aromatic, extraction of nitrated compound with ether, washing with alkali, and treatment of washed ether extract with alkaline butanone solution. A crimson color indicates the presence of benzene. A color which changes to blue on the addition of acetic acid indicates the presence of toluene. Comparisons of these colors with standard colors permit rough quantitative estimations.

TEST VARIABLES. Reproducibility of chemical tests on thinners is determined by the pains taken with the work by the operator. Certain variables affect the solvency and lifting tests, however, which must be controlled before significant results are obtained.

In the kauri-butanol test, the temperature of the solvent and the visual acuity of the operator are important elements. It has been found that the variation of temperature at the end-point does not result in significant changes of turbidity and therefore the process of turbidity might not be considered measureably reversible. This cannot, however, be taken as evidence of lack of temperature effect, since temperature variations during titration are known to exert appreciable influence on the test results (136). In connection with personal errors, the author attempts to eliminate this by allowing one operator to conduct all solvency tests and by constantly checking this point with materials of known kauri-butanol value.

Baldeschweiler, Troeller and Morgan (136) consider an average deviation of 1% in the kauri-butanol test as a maximum precision. The author has found it possible to limit the absolute error easily to 0.5 cc. of thinner titrated.

Lifting action of a solvent depends upon the age of the varnish film to be tested, the presence of moisture in the film, the thickness of film, and the roughness of the surface on which the film is coated. It would appear, however, that lifting time cannot be repeated to closer than 10 to 15 percent.

Sulfonation residue tests can be repeated to within an absolute error of five percent. The modified test (295) using concentrated sulfuric acid usually gives a residue of 95 percent with thinners which are known to contain no aromatics, but this is believed to be due to physical solution of the thinner in the acid, and not chemical reaction.

Tests conducted at the author's laboratory indicate that as little as one percent benzene can be determined by the colorimetric butanone test.

SIGNIFICANCE OF THE TEST. Examination of the recovered volatile from a varnish often indicates what the probable performance of the varnish will be. Also, if the varnish possesses excellent performance characteristics in all respects, but is thinned with a high-solvency coal tar thinner, then it might be possible to use a magnet wire in connection with this varnish that is known to have good resistance to softening by solvents. The test information in this manner, becomes an aid to engineers in designing electrical equipment.

THE VARNISH SPECIFICATION.

A specification is a written attempt to reduce the physical evaluation of a material to some common-ground basis, using methods which best provide a clear definition of the material in the light of the performance requirements as expected in service. This does not mean that actual service conditions should be used in preference to more precise abstract tests if some relationship between the significance of each can be found.

THE FORM OF THE SPECIFICATION. Varnish specifications may take one of two forms: the formula specification and the performance specification. As the name implies formula specifications require a specific mixture to be supplied which experience has shown will be generally satisfactory for the service performance required of it. The objection to this type of specification, however, is that unless considerable research on new materials is constantly undertaken, the specification will eventually describe an obselete or inferior material.

Performance specifications, on the other hand, stress actual service usefulness and usually allow some freedom of choice of ingredients. Notwithstanding a certain amount of development which must be undertaken from time to time in the study of improved testing methods, performance specifications are generally more popular with both customer and supplier.

Some types of specifications require that although the requirements are principally performance in character, that the type of varnish supplied will be the same from shipment to shipment, that is, it will be produced on the same formulation. This is often the case where the testing facilities are limited and where contracts are awarded with acceptance of delivery pending the results of tests. This requirement usually takes the form of a brand or type approval test on a specific, but not necessarily disclosed, formulation. The customer usually holds the supplier to this formulation, since it is reasonable to assume that with the exception of normal manufacturing variations the varnish will be usually acceptable if originally found so. This plan reduces the time spent in testing obviously poor materials and concentrates the effort on varnishes which will probably conform closely to the specification requirements.

Present thought on the brand approval indicate that this procedure will probably give way to a system of statistical quality control. Thus, the brand approval may be a composite test of four to five contract samplings on which a control chart for major performance requirements is drawn (119). Subsequent samplings will be expected to fall within certain control limits, and any adjustments made in the basic formula or processing will undoubtedly be evidenced in the manner in which the test results compare with the previous history of testing.

THE CONTENT OF THE SPECIFICATION. Primarily, the varnish specification should state all of the requirements considered necessary to indicate adequately the service performance of the material, and, secondly, the service which the material is expected to perform should determine the specification requirement values. Diemer (130) advises, in this respect, that the values of the requirements be selected judiciously, preferably aiming at an average material. The values for these average material performance requirements, of course, may be obtained from any source of reliable engineering information. Oftentimes it is advisable to conduct exploratory tests on a large number of commercially-available materials to determine what the industry as a whole can produce with respect to certain major performance requirements.

As an example of this approach, the data shown under Figures 13 and 14 were obtained by examining about 150 insulating varnishes in accordance with Navy Department Specification 52V13e.

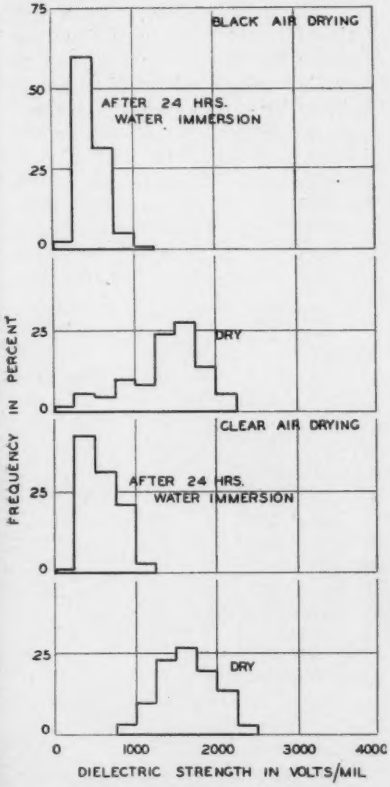


FIGURE 13.—Performance Range of Commercial Insulating Varnishes With Respect to Dielectric Strength Before and After Water Immersion.

The specification should also contain a section wherein is described in every detail the procedure to be followed in determining compliance The procedures selected should embody the best techniques known, and should contain all of the information necessary to conduct the tests, discussing at length the

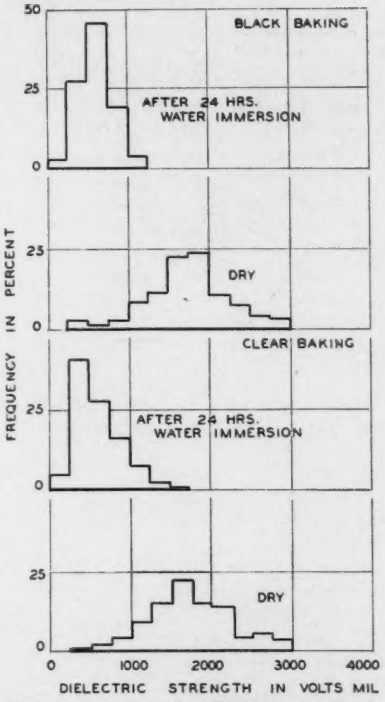


FIGURE 13. (Cont.)—Performance Range of Commercial Insulating Varnishes With Respect to Dielectric Strength Before and After Water Immersion.

effect of varying specified test conditions. With the exception of the workmanship clause, which is falling into increasng disrepute, each requirement should have a corresponding test method.

It is usually difficult for the average consumer to realize the various technical problems concerned with the manufacture and testing of insulating varnish and for that reason the comments of producers should be taken into consideration in the formulation of specification test procedures. Requirements should only be set after a conclusive test program has been completed on all types of varnish. The determination of what the industry as a

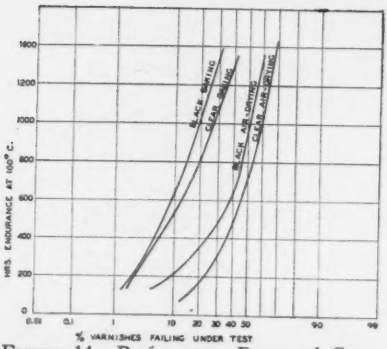


FIGURE 14.—Performance Range of Commercial Insulating Varnishes With Respect to Heat Endurance.

whole can produce, together with what the consumer is willing to pay in premiums for higher-than-average properties of the varnish, will then decide what requirements are eventually incorporated in the specification.

In addition to a complete description of the method of obtaining a test specimen which is representative of the entire shipment, the specification should describe the manner in which retests are to be made. Precision of test results and the degree to which test specimens may fail to conform to requirements are both elements in considering a retest program. The retest method may best take the form of a sequential sampling form as shown in the following table for dielectric strength measurements (Navy Spec. 52V13f):

Dry Dielectric Strength.

Grade	R	A	R	A	R	A	R	A	R	A	R	A
BA	262	772	431	686	487	657	515	643	532	634	599	600
BB	466	1372	766	1219	866	1168	916	1143	946	1128	1066	1067
CA	291	857	479	762	541	730	572	714	591	705	666	667
CB	466	1372	766	1219	866	1168	916	1143	946	1128	1066	1067
CF	291	857	479	762	541	730	572	714	591	705	666	667

Wet Dielectric Strength.

Grade	R	A	R	A	R	A	R	A	R	A	R	A
BA	99	402	188	351	233	334	249	326	259	320	299	300
BB	149	603	299	527	349	501	374	488	389	481	449	450
CA	107	436	216	380	252	362	270	353	281	347	324	325
CB	182	737	366	644	427	612	458	597	476	587	549	550
CF	166	469	233	410	272	390	291	380	303	374	349	350

In the above tabulation, the grades correspond to the designations noted in earlier parts of this work. The symbols R and A denote the volts per mil range of uncertainty, below which the varnish fails (R: rejected) and above which it is accepted (A: accepted). Thus, if the measured wet dielectric strength of a Grade BB varnish based on ten determinations is 220 volts per mil, then a retest of ten additional determinations is made, and the new average calculated on the basis of twenty determinations. If this average is, say, 400 volts per mil, then according to this plan it falls still in the uncertain range of 299 to 527 volts per mil, and a third retest is made. This procedure is followed in this manner until the calculated average either falls outside of the uncertainty range for that particular retest column, or until the sixth retest, at which point the uncertainty range is but one volt per mil, and the varnish either fails or passes finally.

This sequential plan is formulated on data showing what the industry as a whole can produce and what the consumer proposes to assign as a lower limit of tolerance, in addition to the establishment of risks of each party in regard to percentages of good varnishes which will fail the test and of poor varnishes which will pass.

PART TWO

REPRINTED ARTICLES

The articles contained in this Part Two represent those which, in the opinion of the editorial staff, are the most valuable which have been gleaned from recent issues of contemporary publications.

This material was formerly published under the heading "NOTES". It has been given somewhat more prominence and made more readable to enhance its value. The editors' purpose here is to provide a convenient repository to readers of the JOURNAL, for articles originally published elsewhere which contribute to the purpose of the Society; to further the advancement of naval engineering.

AERIAL VIEW OF U. S. S. *VOLADOR* (S. S. 490) *Official U. S. Navy Photograph.*

THE FUTURE OF BRITISH SHIPBUILDING—The Threat of Nationalization: An Analysis.

ACKNOWLEDGMENT.

The following article is reprinted from "The British Motor Ship" for February, 1949 and was written by the editor of that periodical.

Last month the United Society of Boilermakers completed a case for the nationalization of the shipbuilding industry, and this was passed to the Labor Party Executive for consideration, with the hope that it would be put forward in the 1950 election programme of the Labor Party.

It may be said, and truly believed by many who have the welfare of the shipbuilding industry at heart, that nationalization would be so illogical and unwise that there is no danger of its being adopted. Such optimists may, therefore, discount the threat and all that it implies, but they should remember that we have a Government which, in its actions, is sometimes governed more by ideological than logical considerations, as exemplified in the Iron and Steel Bill. For no convincing reply has yet been given to the logical case made out against nationalization of the iron and steel industry. Whatever view the T.U.C. may take of the aspirations of the United Society of Boilermakers, therefore, the shipbuilding industry should not ignore them, at any rate, not entirely. And it should endeavor to keep the public informed.

At a period during the war when the question of standardization was being raised, the writer had correspondence with Mr. Morrison, who made it clear that nationalization as a principle should only be accepted when the industry concerned showed such marked trends towards unification, that it became or was likely to become monopolistic. That principle he has maintained in recent speeches.

No Sign of Monopoly.

There is and can be no trace of monopoly in the British shipbuilding industry in its present structure. There are 30 firms in this country, each of which may be considered as builders of large ships; in addition, well over 20 construct medium-sized vessels. Many, if not most, of them carry on repair work as well, and, in addition, there are a dozen big establishments concentrating wholly and exclusively upon repairs. About 12 large factories are engaged on the building of engines for installations in ocean-going ships, while about one-half of the shipyards have engine-building establishments, and there are substantial numbers of factories where engines for the smaller classes of vessels are built.

The competition between the shipyards has always been severe and with the additional competition in foreign yards between the two wars (not only for foreign orders, but for British ships), the British prices for new tonnage have always been kept down to

AERIAL VIEW OF U. S. S. *VOLADOR* (S. S. 490)

reasonable figures, and it has been necessary to maintain the highest efficiency to meet this national and international competition. No claim can, therefore, be made of anything approaching monopoly in shipbuilding.

Under nationalization, without the spur of competition, and with the interest in his work which it seems to engender in man as he is at present constituted, efficiency would decline and there would not be the same urge to build at the lowest possible prices. British shipowners would, therefore, have to pay more for their ships than their foreign competitors building in yards abroad.

Export Orders of £110 million.

There is abundant proof that the shipbuilding industry is not declining in efficiency. If it were, this would be an argument in favor of its nationalization. During the 10 years prior to the outbreak of war, the average tonnage of vessels launched in this country was in the neighborhood of 760,000, compared with 1,860,000 throughout the world, British yards thus being responsible for 40 per cent of the total production.

Last year, 1,212,698 tons were built in British yards out of a total of 2,481,779 tons for the whole world, namely, 49 per cent, and there is not the least doubt that with practically no addition to the labor force, the British output could have been raised so as to represent some 55 per cent of the world's total had sufficient steel been available.

The orders received for foreign ships is another indication of our competitive status, and therefore the efficiency of British yards. Last year, exports of ships were valued at some £40 million, compared with £6 million in 1938, and to-day 35 per cent of our tonnage under construction is for foreign owners, representing about 750,000 tons gross to the value of £50 million Sterling. We have orders in hand in British yards for foreign owners of not less than 1,500,000 tons gross, the value being in the neighborhood of £110 million. In no other industry are we producing as much as the rest of the world combined, and probably no large industry has increased its exports sixfold since 1938.

There are other strong arguments against nationalization which are not so easy to prove factually, but which those engaged in the industry know are equally effective. Shipbuilding is a pioneering industry and governments are not pioneers. With the allied marine engineering, it has always attracted good brains and the adventurous mind. As a nationalized industry with Civil Service mentality, it will not do so, but will draw the "safety first" character, upon which the greatness of Britain has not been built. For instance, the British oil engine which is most widely used in ships to-day, and is being or will be built in several countries abroad, involved years of development and expenditure of probably several hundred thousand pounds before it achieved success. It is certain that no Government department would have proved thus venturesome in its policy.

The Marine Engineering Industry.

Marine engineering is mentioned because it is difficult to see what line could be taken in regard to this industry if shipbuilding were nationalized, since many of the British shipyards engage upon the building of engines in the same works, under the same management, and with a degree of interconnection that would be impossible to sever. Again, shipbuilding is a generic term, but what is intended in the approach of the United Society of Boilermakers is hull building, because a ship involves the products of scores of other industries which could scarcely be nationalized. In fact, expenditure on the actual ship is probably not more than one-

third of that on the total completed and equipped vessel.

Those who are applying for the nationalization of shipping do not wish to include ship repairing, except as carried on in the yards which build ships. In that case the Government would compete with private enterprise, with all the injustices that are involved, and if ship repairing were wholly excluded from nationalization we should, again, have the nationalization of establishments carrying on nationalized and un-nationalized work with the same staff, and often utilizing material drawn from a common pool. The policy is quite unworkable.

As stated previously, the cost of British ships under nationalization would be more than if private control remained. This has already been well indicated with the small experience of nationalization already gained. In that case British shipping would be penalized, the running costs would be higher than those of foreign competitors, as already noted, and subsidies would probably have to be provided for shipping. The British Government's official views on this subject are given in the White Paper "The Long Term Programme of the United Kingdom," which has just been issued:—

"It is a basic condtion of the full attainment of these shipping objectives that the United Kingdom should be able to offer shipping services throughout the world in free and fair competition, and that there should be no governmental obstruction by discrimination and other measures to their acceptance."

Having plainly expressed such views it would be difficult for the British Government to justify the nationalization of shipbuilding which, in its effect, is equivalent to governmental obstruction by discrimination.

THE ATOMIZATION OF LIQUID FUELS FOR COMBUSTION.

ACKNOWLEDGMENT.

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Following a consideration in some detail of the mechanism, the extent, and basic purpose of atomization, and a reference category by category to the principal types of atomizer available, the pressure jet atomizer is described in detail, attention being called to common faults in design and manufacture. The broad point is then made that liquid-fuel combustion is invariably a vapor-phase process, the function of atomization being primarily that of preparing the less volatile liquid fuels for progressive vaporization in suspension in the turbulent air which is its partner in burning. The relatively great importance in liquid-fuel combustion of the small-size spray droplets is emphasized, and an account is given of the process of combustion of an atomized fuel spray. After a general description of the wax method of particle-size measurement, and the determination of the characteristics of a spray in terms of the parameters n and x in the Rosin-Rammler expression, reference is made to the various factors—including fuel preheating—which affect the quality of atomization.

1 THE PROCESS OF ATOMIZATION.

In effecting the atomization of a liquid fuel, force is applied to the liquid to set it in motion, and generally to constrain it to emerge at some stage of the process as an attenuated film. This film is then disrupted into shreds or filaments, either by the application of other forces or by the persisting balance of energy in the liquid derived from the initial force which promoted its flow. The shreds or filaments then break up into numbers of smaller fragments, which thereupon rapidly assume spherical form under the action of the surface tension of the liquid itself.

Broadly speaking, therefore, the atomization of a liquid fuel involves enforced attenuation, followed by ligament or filament formation and disruption. The disruption of the filaments under the action of surface tension follows the d rule, and results in the production of main droplets and Plateau droplets. Since the filaments themselves vary greatly in diameter, the resulting main droplets and the subsidiary droplets produced from them vary in size correspondingly, and in the aggregate the entire atomized spray is thus composed of droplets of a wide range of sizes.

It must not be assumed, however, that there is a clear line of demarcation between the two phases, namely the attenuation and the filament disruption. The formation of the filaments is due in part to the applied force causing attenuation and partly to surface tension, and there is an interplay of these forces during the atomization process, perhaps also with the intervention of outside influences, as, for example, of the air directed into the spray for combustion purposes.

The process of atomization will continue in this way so long as liquid is fed to the atomizer and subjected to the attenuation process; and the rate of fuel flow may be adjusted, within limits, to suit requirements without affecting the continuity of the process.

2 THE REASON FOR ATOMIZATION.

The fundamental reason why liquid fuels are subjected to atomization is that in the great majority of cases they cannot be properly burnt without. For example, whereas a liquid fuel such as kerosene in a beaker cannot be ignited with a match, when this same liquid is atomized by being passed at a moderate pressure (50 lb./sq. in.) through a simple atomizer a spray is produced, which, on the approach of a match flame, at once ignites and continues to burn by itself.

If the pressure is now materially reduced the spray form changes, becoming coarser and imperfectly developed, and a stage is reached where the liquid will barely ignite, and cannot hold flame by itself.

The reason why it will not burn in bulk, or in coarse droplet form, is that there is insufficient liquid surface exposed to air. Atomization breaks up the bulk liquid into a host of tiny droplets, causing a greatly increased surface area of the liquid to be exposed to the air, and permits of intimate mixing.

3. THE EXTENT OF ATOMIZATION.

Consider unit volume of the liquid (1 cm^3.) in the form of a sphere. The diameter of a sphere of this volume is approximately 1.24 cm. and its surface area 4.83 $cm.^2$. Assume, now, that we break up this 1 $cm.^3$ of liquid into 1,000,000 droplets of equal size. Each droplet will have a volume of 10^{-6} $cm.^3$, and its diameter will be 0.0124 cm. (or 124μ). Thus, the diameter of each of the droplets will be 1/100 of that of the original 1 $cm.^3$ sphere, and the total amount of surface now exposed to air will be 483 $cm.^2$, i.e. 100 times that exposed before atomization. In actual fact, however, a normal spray will contain particles ranging from 5μ up to 500μ in diameter, and each cubic centimeter of liquid will yield nearer 10 million than 1 million droplets; and the surface area of liquid exposed to air may range up to 1,200 $cm.^2$

4. VAPORIZATION.

An alternative method of preparing a liquid fuel for combustion is by heating it above the boiling-point of its heaviest hydrocarbon ingredient, so that it is entirely converted into vapor prior to or while being mixed with air for combustion. This method is, however, applicable only to such high-grade fuels as can be completely vaporized, leaving no solid residue. With the vaporizing burners that have been designed to handle slightly heavier grades of liquid fuel than those just mentioned, it is necessary to scrape clean the strategic parts of the burner at fairly frequent intervals. Historically, vaporizing devices were developed before atomizers; the latter followed, of necessity, as the only effective means of handling the heavier grades of fuel.

5. METHODS OF ATOMIZATION.

5.1. Classification of Types.

Category 1. *Air Atomizers.*

(*a*) Low-pressure air (L.P.A.) burners, in which air, at moderate pressures ranging from, say, 10 in. up to 25 in. w.g., is employed to break up the fuel. Owing to the low pressures involved, at least some 25 per cent of the total air required for combustion is used in the burner for "atomizing" purposes, and it is quite common practice to pass all the air so required through the burner. The Wallsend L.P.A. atomizer (Fig. 1) is of this type.

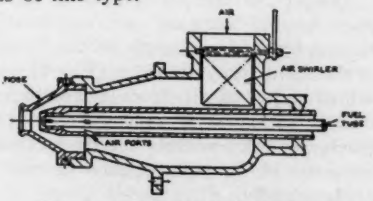


FIG. 1.—*Wallsend L.P.A. atomiser.*

(*b*) Medium-pressure air (M.P.A. burners), in which air at 1½ to 10 lb./sq. in. is employed as the atomizing agent. Usually from 3 to 5 per cent of the air required for combustion is passed through the burner, the rest being supplied as secondary air. The diagrams in Fig. 2 show typical details of this type.

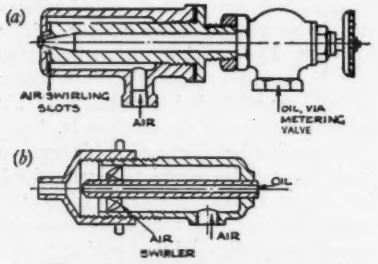


FIG. 2.—*M.P.A. atomisers.*
(*a*) *External mixing.*
(*b*) *Internal mixing.*

(*c*) High-pressure air (H.P.A. burners), in which air at pressures up to 100 lb./sq. in. or more may be employed. In such cases, on grounds of economy, it is rare for more than 2 or 3 per cent of the air to be passed through the burner, the rest being supplied by fan or natural draught as secondary air. The Clyde atomizer (Fig. 3) is an example.

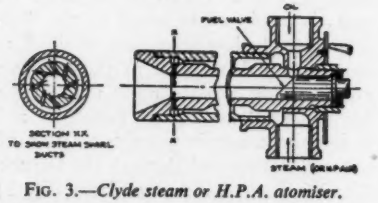


FIG. 3.—*Clyde steam or H.P.A. atomiser.*

Burners in the air atomizing class may be of "internal" or "external" mixing types. An example of the latter is the scent-spray atomizer (Fig. 4), in which a jet of air from a nozzle is caused to shear off and shatter fuel emerging from another nozzle located adjacent to it, and roughly at right angles axially.

Category 2. Steam Atomizers.

(*a*) Medium-pressure steam burners, using steam at 5 to 15 lb./sq. in. gauge. The steam consumption will average about ⅓ lb. per lb. of liquid fuel atomized, representing some 2 to 2½ per cent of the steam that could be generated in a boiler fired by the burner.

(*b*) High-pressure steam burners, using steam at pressures from 20 lb./sq. in. gauge and upwards. Fig. 3 may be taken as an example of this type also. Mixing may be either "internal" or "external," and the scent-spray principle employed, as with air atomizers.

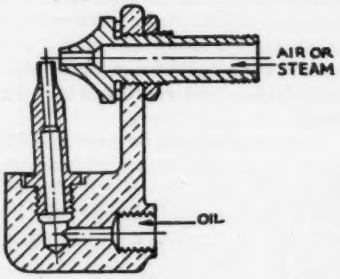


FIG. 4.—Scent-spray atomiser.

Category 3. Drooling or Weir Atomizers (Air or Steam).

The liquid fuel, emerging from a slit, cascades over into a stream of steam or air emerging from another slit immediately below (Fig. 5). The fuel film is sheared through and torn violently into shreds. This type has perhaps a longer history of application than any other, and is still widely used.

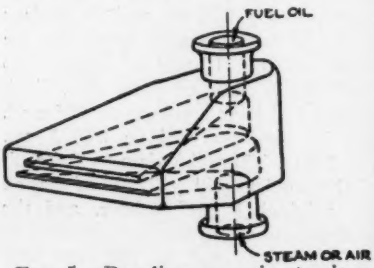


FIG. 5.—Drooling or weir atomiser.

Category 4. *Atomizing by Mechanical Means.*

The liquid fuel is metered at a desired rate into the base of a hollow cup which tapers more or less gradually from base to lip, where the diameter is greatest. The cup is rotated at high speed, so that the fuel spreads over the inner surface as a film and moves forward towards the lip, from which it is flung off by centrifugal force and immediately caught up in an air stream issuing from an annulus surrounding the cup. The axis of the cup may be either horizontal or vertical, and the cup may be rotated by electrical or mechanical means, or by air acting on suitably disposed turbine vanes on the outside of the cup itself, which is then carried in a blast pipe on suitable bearings. Fig. 6 exemplifies this type of atomizer.

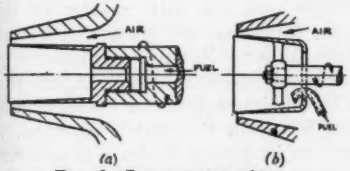


FIG. 6—*Rotary cup atomiser.*
(a) *With shaft feed.* (b) *With pipe feed.*

Category 5. *Pressure Jet Atomizers.*

(*a*) Plain orifice types, as most widely used in diesel and other internal-combustion engines for fuel injection into the cylinders. Commonly these are fed at extremely high pressures (up to 5,000 lb./sq. in. or even higher).

(*b*) Centrifugal swirl types, as used almost universally under the boilers of naval and mercantile vessels, and in large land installations. With these, the fuel pressures required are much lower.

The pressure jet atomizer, of centrifugal swirl type, is perhaps the most widely used of the atomizers described above. It will be described in more detail in Section 5.2.

5.2. Design of Pressure Jet Atomizers: Discussion.

The various types of atomizers mentioned above have been developed to meet different demands and conditions. For example, in a works or factory where steam is abundant, steam atomizers will probably be adopted. Where compressed air is available, high- or medium-pressure air atomizers will be a natural choice. Where there is no supply of either steam or air, but electricity is available to drive a pump or fan, it will be quite natural to adopt the pressure jet atomizer or a low-pressure air atomizer, or else a motor-driven rotary cup type.

Broadly speaking, we may divide atomizers into two classes only: (*a*) those in which use is made of some other medium, such as steam or air, to effect or promote atomization; (*b*) those in which the pressure energy of the fuel itself is used to effect atomization. Rotary cup atomizers (Category 4) come into Class (*a*), since it is the air stream in the annulus around the cup that promotes the attenuation and the atomization of the fuel film centrifugally produced at the lip of the cup by its rotation.

Finally, and again in a broad sense, the mechanism of atomization in these two classes is the same, consisting of a process of attenuated film production followed by ligament growth and disruption, and the ensuing formation of the minute droplets which constitute the spray.

If fuel is forced under pressure through a plain hole or orifice it will emerge as a slender pencil of liquid, and will travel as such for some distance. However, in penetrating the air it becomes more irregular in form, and "necks" develop which lead to its disruption. The surface of the pencil gets rough, small ligaments are thrown off by the surrounding air, and ultimately all the liquid discharged will be found to have been broken up into a collection

of particles or droplets extending over a wide range of sizes. This is atomization, or spray formation. The particular characteristic of this simple form of spray is its high penetrative power, and it finds wide but almost exclusive application in the diesel engine, where the factor of spray penetration is of high importance.

Fig. 7 shows the fundamental features of a simple swirl-type atomizer. The liquid under pressure is fed through tangentially disposed ducts, slots, or channels leading to a circular space called conventionally the vortex chamber or swirl chamber. Admitted tangentially at the periphery of the vortex chamber, the liquid spins or swirls round, its angular velocity varying inversely as the radius of swirl. This implies infinite velocity at the axis of the swirl chamber, which is impossible, and what happens in fact is that an air core is formed, as shown in Fig. 7. The outlet from the vortex chamber is the discharge orifice, which is small compared with the diameter of the swirl chamber, and the rotating mass of liquid is forced forward towards this orifice. It is thus under the influence of two main forces—the translational force which urges it axially forward, and the centrifugal or spinning component which tends to make it fly outwards tangentially directly it emerges from the restricting boundary wall of the orifice. As a result it actually emerges from the orifice as a tube of liquid which is divergently conical and forms a rapidly thinning film. The mechanism of break-up is associated with the rupture of this film due to its instability, whereby it becomes a reticulated network of ligaments, or filaments, which further break up into smaller parts which in turn, under the action of surface tension, form the spherical droplets of various sizes which make up the spray.

The formation of the air core is a characteristic feature of the flow in

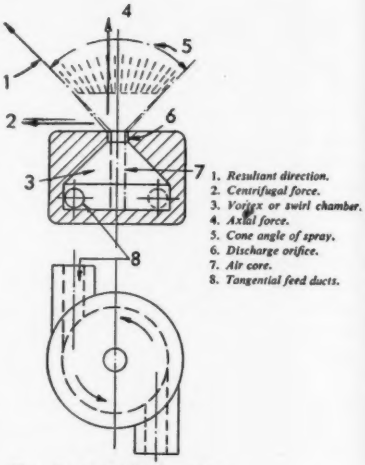


FIG. 7.—*Centrifugal atomiser*

swirl-type atomizers, and accounts for the extremely low coefficients of discharge obtained with nozzles of this type. Measurements show that the air core tends to increase in diameter as the supply pressure increases, and that it is of uniform diameter throughout its length within the vortex chamber. A limit to its growth in size with pressure is imposed by the fact that ultimately, with further increase of supply pressure and hence output, the viscous losses impose an upper limit to the angular velocity, thus preventing further growth, and tending to lead to a slight diminution in spray cone angle as more fuel is forced through the orifice owing to the increasing pressure.

Fig. 8 shows several common forms of swirl-type atomizer. That shown in Fig. 8 (*a*) was widely used for a decade before World War II. It consists of two parts—a body containing the final orifice, and a plug member in the truncated conical tip of which are cut the two, three, four, or more tangential slots. The conical inner surface of the body serves to roof-in these slots. Fluid under pressure entering the body from the supply pipe can find egress only

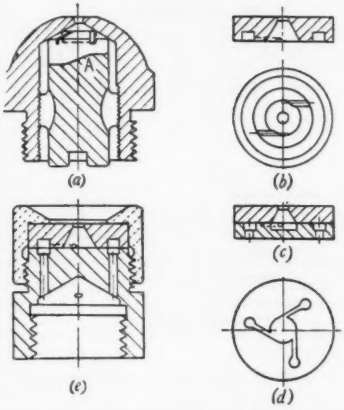


FIG. 8.—*Types of simple swirl atomiser.*

(a) Monarch type.
(b) Disc or plate type (single).
(c) Two-disc or plate type.
(d) Stamped slot plate.
(e) Holder or housing for (b), (c) and (d).

after passing up the tangential slots and swirling in the vortex chamber. This type of atomizer is made in a standard size externally, but in capacities of 1 gal./hr. up to 50 gal./hr. In Fig. 8, (*b*), (*c*) and (*d*) show three forms of disc- or plate-type atomizer any of which can be accommodated in a simple housing consisting of a body and cap, as indicated in (*e*). A single flat disc contains all the essential parts of the atomizer shown in (*b*), viz. tangential slots, vortex chamber and orifice, the flat top of the housing or body serving to provide the back wall of the vortex chamber, and to close the tangential slots along their length.

The design shown in Fig. 8 (*c*) is similar except that the slots are cut in one plate or disc and the orifice and vortex chamber in the other. Fig. 8(*d*) shows another form of two-disc assembly, the slotted member being punched out or stamped. An advantage of the two-plate design is that different forms of slot plate can be used with a particular orifice plate, and vice versa. It is essential that the two discs be in exact coaxial alignment.

Fig. 9(*a*) shows another form of construction in which the feed holes are drilled through the wall of the vortex chamber tangentially. In the smaller sizes, particularly, the drilling operation may present difficulties.

Two other types are shown in (*b*) and (*c*), Fig. 9, the former having multiple-start grooves cut in a cylindrical member which is a close push fit in the cylindrical bore of the orifice component, and the latter with spiral grooves on a conical member which fits in an extension of the vortex-chamber cone. In this latter type the spiral grooves may be large at entry, and diminish to a minimum size at their exit to the swirl chamber. A variant of the Fig. 9(*b*) form is also shown, in which two oblique ducts are drilled in the cylindrical plug (Fig. 9(*d*).

Fig. 9(*e*) is an example of bad design. The first fault is that the spirally

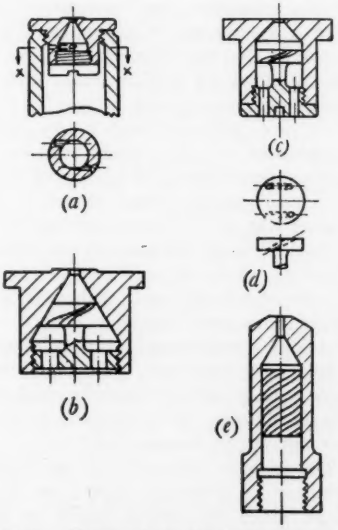


FIG. 9.—*Types of simple swirl atomiser.*

(a) Drilled tangential-duct type.
(b) Spiral grooves in conical plug.
(c) Spiral-groove type.
(d) Alternative with ducts drilled obliquely.
(e) Excessively long spiral grooves and orifice.

grooved plug requires the fuel to traverse too long a path before being discharged into the vortex chamber, with the result that there is an unnecessary pressure loss. The second fault is that the discharge orifice is too long, and consequently the spray cone angle is reduced.

In the centrifugal atomizer the ideal orifice would be knife-edged, but it is not a practical proposition to make a hole of a precise size knife-edged, and further, such a hole would rapidly become enlarged by the erosive action of the swirling fluid passing through it. Usually, the final orifice of a swirl-type atomizer is made about ¼ to ½ a diameter long, and rarely over 1 diameter long.

Fig. 10 illustrates some not uncommon faults in orifices. In (*a*) the orifice is oblique to the axis of the vortex chamber, and in (*b*) it is eccentric to the vortex chamber. Case (*c*) is perhaps less common; here the orifice and vortex chamber are concentric but both are eccentric to the outside diameter of the disc. The use and assembly of such an orifice plate with a slotted companion plate, where the two discs are held concentric by their peripheries, would clearly lead to a serious eccentricity of the swirl slots in relation to the vortex chamber and orifice. Fig. 10(*d*) shows the case of a ragged finish at the apex of the vortex chamber where the orifice starts, and at the discharge end. Whereas the illustrations show palpable errors, eccentricities and obliquities, which could hardly pass unnoticed, it is the minute error, the minute eccentricity, and the minute obliquity which impair the performance of an atomizer.

The simple pressure-jet swirl-type atomizer is severely limited in flow range, since the output varies as the square root of the fuel supply pressure, i.e. $Q \propto P^{0.5}$. Assuming, for example, that a modest output range of, say, 4 to 1 is required in certain application,

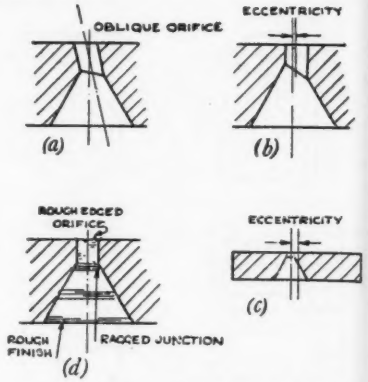


FIG. 10.—*Faults in orifices.*

and that the minimum operating pressure for effective atomization of the fuel by the atomizer is 60 lb./sq. in., the pressure required to quadruple the output would be 960 lb./sq. in.

With steam or air atomizers, a range of 5 to 1 is easily covered without abnormally high pressure requirements; but the intrinsic merits of the pressure jet atomizer in many applications have led to the development of special types with which a greatly expanded range of output is obtainable.

Turning to a consideration of wide-range pressure jet atomizers, the oldest of the class is the spill-type or Peabody-Fisher atomizer. The fuel flowing through the tengential swirl-producing

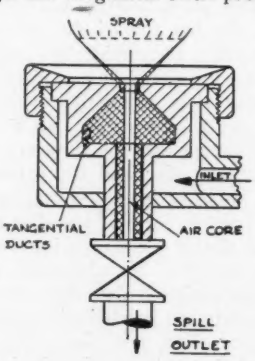


FIG. 11.—*Features of spill-type atomiser.*

slots has two exit paths from the vortex chamber (Fig. 11), namely the normal outlet orifice to the burning space and another orifice concentric with it but in the rear wall of the vortex chamber, and generally of larger size. This is the spill orifice, which is ducted back by a pipe, in which there is a control or throttle valve, to the fuel tank. It will be readily perceived that if the valve in the spill line is fully open the greater part of the fuel which flows through the tangential ducts will be bled away back to the tank, and only a small proportion will pass out of the final orifice. Indeed, it is possible to arrange the relative sizes of the discharge and spill orifices so that all the fuel passes back to tank and none emerges from the final orifice, which in this case must simply be of the same size as, or smaller in diameter than, the air core.

Having thus said, in effect, that zero output is possible, it is not difficult to see that as between spill wide open (zero effective output) and spill closed (all flow through final orifice) a wide range can be obtained even at one fixed supply pressure; and when pressure regulation is coupled with spill throttle-valve control a still wider range can be covered. In actual fact, using a constant supply pressure, a range of 20 to 1 can be quite easily obtained by spill control alone, and if the pressure were varied over a 25 to 1 range than a burner range of 100 to 1 would be obtained. The spilling-type atomizer has many attractive features, and involves no moving parts, but like other types

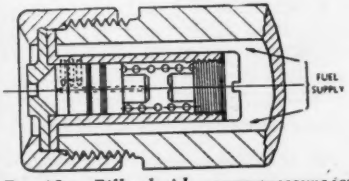


FIG. 12.—*Pillard wide-range pressure jet.*
As piston recedes with rising fuel pressure, additional swirl ports become operative.

of wide-range atomizer to be described it gives a markedly diminishing spray cone angle with increase of output.

The main feature of the Pillard atomizer (Fig. 12) is the provision of variable area in the tangential swirl slots, which, in conjunction with a modest range of fuel-pressure adjustment, enables an output range of 10 to 1 to be obtained.

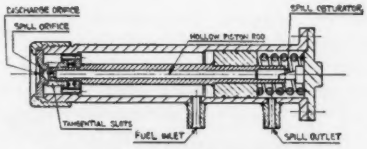


FIG. 13.—*Lubbock wide-range burner.*

The Lubbock atomizer (Fig. 13) combines the spill feature with the Pillard feature; the servo-piston, responding to rising fuel pressure, automatically increases the tangential slot area and at the same time throttles the spill. Ranges up to 50 to 1 in output can be obtained with this type.

The Duplex (Figs. 14 and 15) is a split-feed atomizer having two sets of tangential slots, namely small primary slots and large secondary slots with separate feed passages. The illustrations show a form developed by Joseph Lucas, Ltd., for application to the Rolls-Royce Nene turbo-jet engine. In operation, fuel is fed at relatively low pressure to the primary slots only, and at some appointed stage of rising pressure is admitted to the secondary slots also, materially augmenting the fuel output.

The Todd (Fig. 16) is a spill-type atomizer which has the unique feature of pruning off a varying proportion of the fuel through a split in the length of the discharge orifice.

6. THE COMBUSTION OF LIQUID FUELS.

It is sometimes claimed that a particular atomizer gives a spray composed of droplets of uniform size, and, further, that such spray is ideal from a combustion point of view.

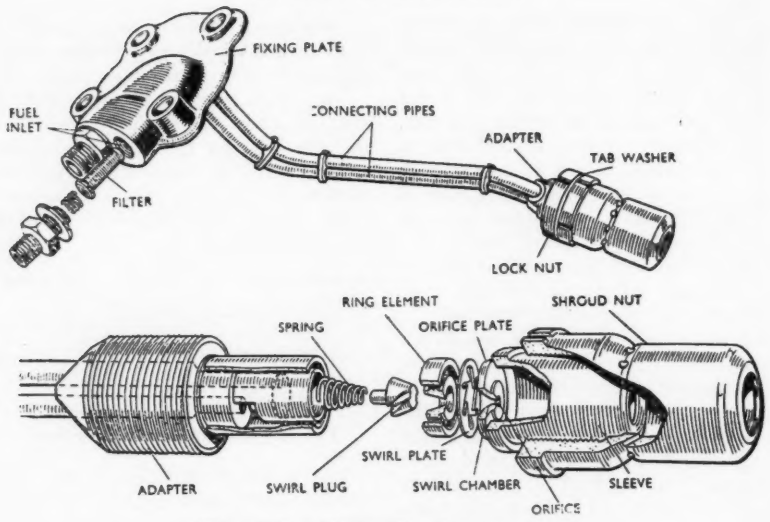


FIG. 14.—*Lucas Duplex atomiser.*

The first of these statements is disproved by the systematic work which has been carried out on the structure of fuel sprays, and to which reference will be made later in the paper. There may, however, be some truth in the second statement providing it is made clear that there is an upper limit of size of droplet.

The initiation (i.e. intial ignition) and continuation of the combustion of a liquid fuel spray depend upon several factors, of which the efficient atomization of the fuel is certainly the first and probably the most important. The simple experiment described above, in which stable burning was obtained when the atomization was good, and in which the flame could not sustain itself when the atomization was poor, supports this view. Systematic research reveals that, with a pressure jet atomi-

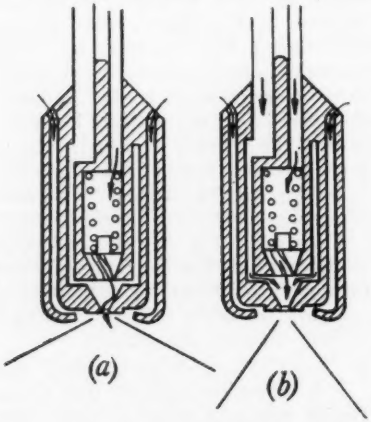


FIG. 15.—*Lucas Duplex atomiser.*
(a) Primary flow.
(b) Duplex flow.

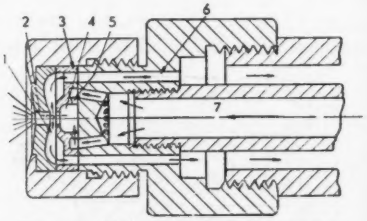


FIG. 16.—*Todd variable-capacity atomiser.*
1. *Oil-return inlet.*
2. *Orifice.*
3. *Sprayer plate.*
4. *Whirling chamber.*
5. *Tangential slots.*
6. *Oil-return passage.*
7. *Fuel supply.*

zer, for example, when operating at low pressures the spray structure is more uniform in droplet size than at high pressures. The mean droplet size in the low-pressure case is greater, however, than in the high-pressure case, although in the latter a greater range of droplet sizes is produced. The fact that a considerable proportion of the spray is then composed of droplets in the extremely small size range (25μ and under) assists the initial ignition and promotes the stable continuity of burning. Once the flame is established it is almost certain that many of the smallest droplets successively emerging from the atomizer are completely evaporated before burning, and provide a stable gaseous flame region as a foundation for the rest of the flame.

The small droplets in a spray are extremely important, since they ignite readily, and, if sufficiently numerous, provide the basis for the chain reaction upon which continuous stable burning of the rest of the spray depends.

The proportion of big droplets that can be tolerated is dependent upon the relative proportion of very small droplets, since the flame-sustaining capacity of the latter will affect the rate of burning of the former. A big droplet burns away progressively in its turbulent flight path through the air in the combustion region, the vapor envelope being continuously replenished from the residual liquid of the diminishing droplet until the supply is exhausted. Very big droplets may not be completely consumed, and may pass out as liquid in the exhaust gases, to the debit of the heat balance. They may, on the other hand, strike some part of the furnace structure and cause a deposit of hard carbon to be gradually built up.

In general, therefore, large droplets of liquid fuel are to be avoided as far as possible unless there is adequate time for their complete combustion, e.g. in an oil-fired steel furnace, where their flight path is long and the ambient-temperature conditions favor their complete combustion. Indeed, in an oil-fired steel furnace the presence of large droplets may be desirable in that they increase the burning time of the fuel, so that the entire bath benefits from the firing.

In burning liquid fuels it is always necessary to provide more than the amount of air theoretically needed for complete combustion, and the art of efficient liquid-fuel burning lies largely in keeping the excess-air quantity to a minimum. Thus combustion efficiency can be expressed as:—

$$\frac{\text{Quantity of air required for chemical reaction combustion}}{\text{Quantity of air actually used to secure complete combustion}} \times 100 \text{ per cent}$$

An alternative measure of combustion efficiency is the total heat liberated expressed as a percentage of the calorific value of the fuel.

In oil-fired applications generally, the combustible mixture of atomized liquid fuel and air is ignited locally by means of a spark or torch, which initiates a combustion wave that is propagated through the entire mixture. The combustion wave moves against the mixture stream, and flame stability depends upon balancing the velocities of the combustion wave and of the mixture stream. Under certain conditions of fluctuating stream velocity the flame front will oscillate and a pulsating or hunting flame will result.

Stability of flame in combustion is achieved when the mixture stream velocity and the velocity of flame propagation are equal; and the flame front is then established over some area, ideally in a plane normal to the stream axis, where this condition of velocity balance exists.

The work of the late Prof. W. T. David showed that flame propagation after ignition is stimulated by the trans-

fer by diffusion of activated particles into the unignited region, and this is subsequently assisted by the thermal radiation from the flame; the whole process depends upon the presence in the oil spray of a sufficient proportion of very small droplets. In the process of burning a liquid fuel there is a series of transitional steps, involving intermediate dissociation products which react at high energy-levels, between the original fuel/air mixture and the final carbon dioxide and water which pass out of the chimney to waste. These phenomena were described recently by Dr. R. Pendennis Wallis in a paper presented to the Liverpool Engineering Society.

7. The Structure of Atomized Sprays.

Investigations concerned with the structure of atomized sprays rely upon measurements of the size distribution in particulate materials, a field in which much painstaking work has been done both in this country and abroad.

During World War II a technique was developed for the measurement of the structure of liquid sprays in connection with research on fuel atomizers for the Whittle jet engine. This technique was based upon the observation that paraffin wax, when heated to a suitable temperature level above its melting-point, corresponds closely in the significant characteristics of viscosity and surface tension with kerosene, the fuel used in the jet engine.

In making droplet size tests, the atomizer is fed with liquid paraffin wax, and produces a spray to all appearances exactly like that produced with kerosene. The minute liquid wax droplets solidify rapidly in the air, and can be collected on glass slides for detailed examination, measurement and classification.

In the technique as later developed, the entire wax spray is directed into a funnel-like bath of flowing water (Fig. 17) and a sample of the water-borne spray is collected in a large beaker. Generally between 5 and 10g of wax droplets are collected, and the sample is then passed through a series of six or more graded gauze sieves, the smallest particles which pass through all the gauzes being finally trapped on the surface of a filter paper. The weight of wax droplets collected on each gauze and the filter paper is then determined, and in this way the spray distribution by weight in a set of known size groups is obtained. From a plot of the data it is possible to determine the characteristics of particular sprays.

The accompanying Table shows a typical set of test results relating to a single atomizer operated at four different fuel supply pressures. The values of

83° Pressure jet atomiser : capacity 81 lb. fuel per hour at 100 lb./sq. in.

Fuel pressure, lb./sq. in.	n	$\bar{x}$, in μ	Specific surface, $cm.^2/cm.^3$	Surface mean diam., μ	50% size, μ	% By weight of spray in droplets	
						> 225μ	> 275μ
12½	2.8	225	375	160	197	37	19
25	2.6	190	462	130	165	21	8
75	2.3	143	659	91	122	5.6	1.2
125	2.2	121	790	76	102	1.5	<0.1

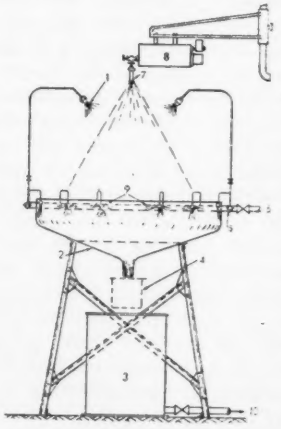


Fig. 17.—*Collecting funnel for wax.*

1. *Water sprays used to keep fine particles from rising (used additionally to the water jets in the funnel).*
2. *Vitreous-enamelled steel funnel, 4 ft. diam.*
3. *Sump tank.*
4. *Sampling vessel held here, to collect the sample for filtration.*
5. *Ring main.*
6. *Mains water.*
7. *Nozzle on test.*
8. *Wax vessel.*
9. *Water jets.*
10. *To drain.*

n and $\bar{x}$ are derived from the Rosin-Rammler law; this is a general analytical relation for the size distribution of particles in powdered coal and other pulverized materials, but it applies also to the droplet distribution in atomized liquid sprays.

As applied to fuel sprays the law takes the form

$$R = 100e^{-(x/\bar{x})^n} \qquad (1)$$

where R is the percentage by weight of the total spray composed of droplets greater in diameter than x, $\bar{x}$ is the "absolute size constant," and n is the homogeneity coefficient (or distribution constant).

The expression can be rearranged as follows:—

$$\frac{100}{R} = e^{(x/\bar{x})^n} \qquad (2)$$

Passing to logarithms to the base 10,

$$\log \log \frac{100}{R} = n \log x - n \log \bar{x} + \log \log e$$
$$= n \log x - k \qquad (3)$$

If, therefore, $\log 100/R$ is plotted against x on logarithmic paper a straight line is obtained, and the value of n is the slope of this line (i.e. the tangent of the angle it makes with the axis of x).

The value of $\bar{x}$ is the value of x for which

$$\log \frac{100}{R} = \log e = 0.4343$$

If, in the original expression (1) we put $\bar{x} = x$, then

$$R = \frac{100}{e} = 36.79 \text{ per cent}$$

i.e. 36.79 per cent by weight of the spray is composed of droplets exceeding $\bar{x}$ in diameter.

The median droplet diameter, x_m, is the size of droplet above and below which 50 per cent by weight of the spray is found. It can be derived from the expression

$$x_m = \bar{x} (\log_e 2)^{1/n} \qquad (4)$$

It is, however, more readily determined from the straight-line graph by finding the value of x for which

$$\log \frac{100}{R} = \log 2 = 0.3010$$

The *surface mean diameter* (S.M.D.) is the droplet size in a hypothetical uniform spray having the same surface area per unit volume as the actual spray. It is complementary to another frequently quoted spray characteristic, the *specific surface*, which is the actual superficial area produced by the atomization per unit volume of liquid sprayed.

The specific surface of the spray can be determined once the values of x and n relating to the spray have been found. It is derived as follows:—

The number of droplets of diameter $x \rightarrow x + dx$ per unit volume of fuel is given by

$$Ndx = \frac{6 \times 10^{12} n}{\pi \bar{x}^n} \cdot x^{n-4} \cdot e^{-(x/\bar{x})^n} dx$$

$$\therefore \text{ Specific surface} = \frac{\int_0^\infty x^2 N dx}{\int_0^\infty 1/6 x^3 N dx} \text{ square microns per cubic micron}$$

$$= 6 \times 10^4 \cdot \frac{\int_0^\infty x^{n-2} \cdot e^{-(x/\bar{x})^n} \cdot dx}{\int_0^\infty x^{n-1} \cdot e^{-(x/\bar{x})^n} \cdot dx} \text{ cm.}^2 \text{ per cm.}^3$$

$$= \frac{6 \times 10^4}{\bar{x}} \prod \left(1 - \frac{1}{n}\right)$$

That is,

$$\text{Specific surface } (S_o) = \frac{6 \times 10^4}{\bar{x}} \cdot \frac{\prod \left(2 - \frac{1}{n}\right)}{\left(1 - \frac{1}{n}\right)} \text{ cm.}^2 \text{ per cm.}^3 \quad . \quad . \quad (5)$$

The surface mean diameter (S.M.D.) may be derived from the specific surface value (S_o) by making use of the following relationship:—

$$\text{S.M.D.} = \frac{6 \times 10^4}{S_o} \text{ microns} \quad . \quad . \quad . \quad . \quad (6)$$

The surface mean diameter (S.M.D.) may be derived from the specific surface value (S_o) by making use of the following relationship:—

$$\text{S.M.D.} = \frac{6 \times 10^4}{S_o} \text{ microns} \qquad (6)$$

The S.M.D. is thus directly proportional to the value of $\bar{x}$. It is sometimes called alternatively the "Sauter" mean diameter, after J. Sauter (U.S.A.), who put it forward as a significant index of atomization quality in his work on diesel jets.

Thus by applying the Rosin-Rammler law to liquid fuel sprays it is possible to describe the droplet distribution in

terms of the two parameters $\bar{x}$ and n, and thence to define various other characteristics of the spray.

Principal among these are the *specific surface*, the *surface mean diameter* and the *median diameter*. Furthermore, the graph of the weight distribution gives the value of R (percentage by weight oversize x) for any value of x, as well as, for example, the order of spray volume (or weight) in the large size end of the scale (say, $> 250\mu$ diameter).

The index n in the Rosin-Rammler expression is called the homogeneity coefficient, or distribution constant. In the Rosin-Rammler law

$$R = 100_e^{(x/\bar{x})^n}$$

and in the limiting case where $n = 0$ the standard deviation (σ) of the distribution is infinity, i.e. droplets of all sizes occur. Alternatively, if n is infinite σ is zero, i.e. the droplets in the spray are all of the same size. Hence the higher the value of n the more uniform the spray.

It is found that for liquid fuel sprays the value of n generally lies between 2 and 4, the higher values of n being usually associated with operation at low-level non-optimum conditions. Hence, whereas under these conditions the spray is more homogeneous, its mean size is high, and the relative value of $\bar{x}$ tends to be high also.

8 Factors Affecting Atomization Efficiency.

8.1 Mechanical Factors.

These include (*a*) the type of atomizer—whether air or steam, rotary cup or pressure jet; (*b*) the scale (physical dimensions) of the atomizer, as determined by its duty.

The investigations carried out to date have shown that, as might have been expected, (*b*) is of great significance.

In the simple pressure jet atomizer the mean size of droplet in the spray varies approximately as $\sqrt{d}$, where d is the diameter of the discharge orifice. Little has so far been done to establish the corresponding approximate quantitative relationship for atomizers of other types, principally because so many more variables are involved.

Such work as has been done on air atomizers indicates that the mean droplet size is appreciably smaller than with the simple pressure jet of equal output, and the same is almost certainly true of steam atomizers. With air and steam atomizers, however, the running costs per ton of fuel atomized are considerably higher than with the pressure jet type.

8.2. Operational Factors.

These comprise the conditions of atomizer operation, such as air or steam pressure, speed of cup rotation, or fuel pressure employed.

Such quantitative information as is available relates to the simple pressure jet, of which the mean spray droplet size appears to vary inversely as $P^{0.35}$, where P is the applied fuel pressure in pounds per square inch gauge.

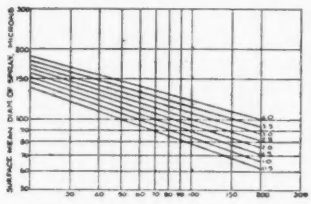


FIG. 18.—*Effect of pressure on mean droplet size, for pressure jet (swirl-type) atomisers.*

Fig. 18 is a family of graphs based on data obtained from over 100 different tests, in which the effect of fuel pressure (P) upon surface mean diameter is shown for atomizers ranging in capacity from 5 to 40 gal./hr.

8.3. Fuel Factors.

These comprise the viscosity, surface tension and density of the fuel, the significant factor being viscosity. Surface tension in hydrocarbon liquid fuels varies only slightly, and has little effect on the quality of atomization. Density may vary from, say, 0.7

(volatile spirits) up to >1 for fuels from particular sources. In general, however, for the industrial and commercial range of fuels the density of the fuel *in the atomizer* will ordinarily vary only from, say 0.85 to 0.9.

Liquid fuels range from grades which flow as freely as water (e.g. motor spirit) to grades as resistant to flow as pitch. Atomization of the low-viscosity fuels (kerosenes, gas oils and diesel oils) is readily effected at atmospheric temperatures without the need for abnormal supply pressures, or air or steam pressures, but more viscous grades have to be preheated so as to reduce the viscosity to the point where the process of attenuation, filament production, and droplet formation can proceed. At some level of viscosity, or reluctance to flow, no amount of applied force will constrain the liquid to break up into a spray of droplets. It will simply ooze through the atomizer rather like toothpaste coming from its tube.

With pressure jet atomizers of small output (say, up to 50 lb. fuel per hr.), satisfactory atomization at reasonable fuel pressures cannot be obtained with fuels whose viscosities exceed 70″ Redwood I (about 15 centistokes). With higher-output atomizers of this type the viscosity limit for satisfactory atomization may be raised a little, say to 100″ Redwood I (= 23 centistokes), providing sufficiently high fuel pressures are available, but it is generally simpler and better to preheat the fuel. Light fuel oil (200″), Navy fuel (400″) and all other grades up to Bunker C grade (6,000″) cannot be atomized efficiently without preheating.

The use of compressed-air or steam atomizers may slightly extend the scope for *cold* atomization. Claims are sometimes made that such and such a burner will satisfactorily atomize cold 200″ fuel; but it is rare to find such claims fully justified, and then only because the conditions of application are favorable, with unusually careful attention given to air supply and mixing, and the existence, in a specially designed combustion chamber of ample size, of

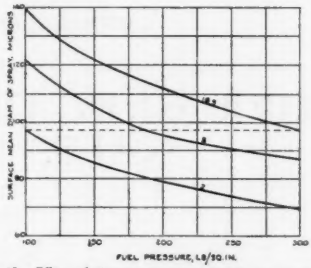


FIG. 19.—*Effect of viscosity on mean droplet size (swirl-type atomiser).*
Numbers against curves are viscosities in centistokes.

suitable refractory surfaces. In the vast majority of applications it is more economical to preheat the fuel than to try to atomize it cold by using more power in the form of greatly increased fuel pressures, higher air pressures or more steam.

Work is in progress to determine the quantitative effect of fuel viscosity as a factor affecting the atomization process, and some preliminary results are shown in Fig. 19. The dotted line drawn to intersect the curves at a common surface mean diameter of 97μ shows that whereas at a viscosity level of 2 centistokes this mean droplet size is obtained with a supply pressure of 100 lb./sq. in., the corresponding pressures at 8 and 18.5 centistokes are 170 and 300 lb./sq. in. respectively.

From this set of curves it is easy to observe the shift in mean droplet size for each viscosity level by noting the intersections with any pressure ordinate of the three viscosity curves.

Satisfactory atomization calls for the use of an atomizer of good design, robust construction and which has been correctly assembled. Given a good atomizer, and a supply of liquid fuel properly conditioned for atomization, it remains to ensure a balanced supply of the correct quantity of air for combustion, and a combustion space adequate for the complete burning of fuel at the rate required for the job. The proper mixing of air with the atomized spray is also a vital consideration in successful oil burning, particularly where the maximum flame temperature is required.

CORROSION OF TURBINE JOURNALS.

ACKNOWLEDGMENT.

This article by MR. S. E. BOWREY is reprinted from the "Institute of Marine Engineers Transactions" Vol. LXI, 1949. The author proposes the use of a water-miscible inhibitor in lubricating oils which have been contaminated by sea water.

INTRODUCTION.

Normal corrosion risks in turbine lubricating systems are already well controlled. The measures herein described were devised for emergency use, rather than for continuous application, to deal with cases of intrusion of sea-water where the corrosion was so bad that machinery was being scrapped as a result. It remains to be seen whether there is any field for them in less extraordinary circumstances.

The infrequency of severe corrosion of turbine journals and gearing has resulted in a scarcity of co-ordinated information regarding it, the circumstances surrounding those cases which did arise often being obscure. When something approaching an epidemic occurred in ship's turbines at the crucial period of anti-submarine warfare, many different explanations were offered and some time passed before the situation became clear. An account of four cases arising in anti-submarine ships is given to illustrate this, and to establish the fundamental facts of a position which could easily recur in a future emergency. Consideration is then given to corrosion-inhibiting oils and to the water-soluble inhibitor, sodium nitrite, herein proposed. An account of the laboratory investigations and sea trials of sodium nitrite follows, with details of successful application of the proposals to the case of a ship where the gearing was already in an advanced state of corrosion.

WAR-TIME CASES OF CORROSION IN MARINE TURBINE.

Case I.

This ship was one of the fifty destroyers transferred to Britain by U.S.A. All her turbine and gearing journals had previously been stoned bright, but soon after she was commissioned in the Royal Navy large amounts of ferric oxide were noticed in the bowl of the centrifugal separator, and a main thrust bearing failed. All journals were found in a state of active corrosion, deeply etched and covered with black oxide which turned red on the surface after opening up. The whole forced lubrication system, including pumps, coolers, storage and drain tanks was dismantled and cleaned, and the journals stoned bright. After reassembly, oil was circulated for forty-eight hours. Inspection showed that journal corrosion had already begun again, but the operational need was such that no further delay could be tolerated and the ship went to sea. She remained in service for eight months, and at the end of that time the bearings showed approximately 0.020-inch wear-down and a further re-fit was ordered. Corrosion was even worse than before. The cleaning of the lubrication system was repeated and in addition extra handholes were cut in the turbine bearing sumps, which were then washed out with carbon tetrachloride. Most of the journals had to be re-machined, and the ship was in dock for three months.

Only eight months later the trouble had recurred and all main journals had once more to be machined. Each time the ship made a short trial run the journals showed fresh rust and had to be cleaned up. At this point several independent opinions were called for and at least six different explanations were suggested, The eventual discovery and repair of a salt-water leak from the bilges into the oil system apparently cured the trouble, no further reports being received. As far as available information goes, a plain mineral lubricating oil was used throughout. Reports on its condition were satisfactory as might be expected from the fact that consumption was heavy enough to equal a complete fill-up in ten days steaming, but curiously enough there is only one mention of any test for salt water having been made prior to the last investigation. This occurs in a report by the Anglo-Iranian Oil Co. The general view appeared to be that such excessive corrosion could scarcely be attributable merely to sea water, and several experts continued to emphasize other explanations.

A very clear appreciation of the part played by salt water appears in a report by the Engineer Officer which includes an account of similar trouble which arose later in a new ship and was caused by connecting the sea water side of the oil coolers to the drain tanks when the ship was under construction.

Case II.

This concerned another destroyer under refit. The condition of her machinery was described in terms which coincided closely with Case I, but the refit was nearing completion and no first hand investigation could be made. Two independent analyses of the emulsified oil and water from the circulation system had been carried out and each report indicated the presence of salt. Each report also showed contamination of the oil with fatty matter, and while ignoring the salt, expressed the view that the presence of fatty matter had caused the trouble. This view arose, no doubt, from the fact that the condition of the machinery first came to light through blockage of oil strainers ascribed by the engine-room staff to the sudden formation of an emulsion. The detection of fatty matter capable of acting as an emulsifier thus appeared to account for everything, the rusting being put down to the circulation of an abnormal proportion of water with the oil. In the light of the experiments recorded below it now appears likely that rusting occurred first, and that the blockage of strainers was due to solid rust. There was, however, no possibility of resolving doubts in either direction, owing to the speed with which repairs had gone forward and the lack of facilities for further investigation.

Case III.

Within a week or two a third entirely similar case arose, again in a destroyer operating in the North Atlantic. The oil in use was in good condition and although emulsified with water in the turbine system, was found when clarified to have an Institute of Petroleum demulsification value of 120, which is quite satisfactory. Rust was present and the water contained 0.05 per cent NaCl, equal to about 20 grains of chlorine per gallon, or 1.5 per cent of sea-water.

This concentration of salt does not appear particularly alarming, but a more significant result was obtained by examining the rusty steel turnings removed when machining one of the journals. On analysis these turnings yielded approximately 1.6 per cent of water-soluble solids consisting of salt and ferric chloride, which leaves little room for doubt that the rust was produced by salt water. With regard to the low concentration of salt found on testing the water, it should be borne in mind that inward leakage of sea-

water would usually be stopped by the outward oil pressure when the engines were started. Meanwhile any sea-water which entered while the engines were at rest would be removed by the separator and slowly replaced by fresh water from the turbine glands. Hence a salinity test made after several days of steaming might fail to indicate the seriousness of a leak, though the loss of oil ought to arouse suspicion.

The very similar conditions found in these three ships appeared to warrant the conclusion that salt water leakage lay at the root of most if not all cases of this type of rusting. The conclusion was reinforced by the descriptions of similar occurrences compiled by the Engineer in Chief's department, the presence of sea water being mentioned in a number of instances. It was considered, therefore, that a successful method of stopping salt-water corrosion would deal adequately with the trouble and a recommendation was made to the effect that periodical testing for salinity of the water discharged by the separator should become a routine precaution.

Case IV.

This arose three years later, when the cause of the trouble was no longer in doubt. The ship was a recently built frigate and depthcharge concussions had so strained her hull that within her first year replacement of a large proportion of her underwater rivets had been necessary. After the war she was laid up for some months and when re-commissioned experienced engine trouble during her first full-power trial. The first sign was the blowing of a gland packing due to excessive wear-down of a turbine bearing. Then it was observed that the oil contained rust, and a test revealed that the water present in the oil system was 90 per cent sea-water. On opening up, heavy corrosion of gears and journals was discovered together with wear-down of bearings. Part of one of the main gear wheels is shown in Fig. 1. The most seriously damaged bearing had suffered 0.065 inch wear-down, which had resulted in rubbing of the turbine rotor against the casing. Although only three years old, the main gearing had to be scrapped. The intrusion of sea-water was traced to two split tubes in an oil cooler.

A point of interest is that for some months before the ship was laid up the oil in use was corrosion-inhibiting turbine oil to U. S. Navy Symbol 2190T. The corrosion therefore took place in spite of the protection of this oil, and it is fair to conclude that however valuable it may be in normal circumstances it is inadequate in presence of a serious sea-water leakage. This was confirmed by trials described on p. 676.

Explanations Put Forward During Discussion of Corrosion of Turbine Bearings.

Before leaving these typical cases it will perhaps be of interest to mention some of the alternative explanations offered, and then to give reasons for regarding salt intrusion as the main factor. The following four explanations of the corrosion (or variations of them) were advanced from one quarter or another.

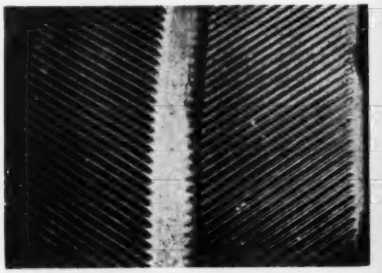


Fig. 1.—Heavily corroded main gear wheel.

(*1*) *Introduction of emulsifying agents in error or by saboteurs.*

There is no clear evidence that the presence of fresh water, even when emulsified with the oil in large amounts, causes rusting to the extent

found, and some at least of the bad cases arose with oil of good demulsification properties.

(2) *War conditions.*

These included the necessity for closing engine-room hatches at night (thus creating an exceptionally hot moist atmosphere), the use of high speeds and sudden changes of course in all weathers (bringing sea-water in at fan intakes), curtailment of opportunities for maintenance work and other adverse circumstances.

These conditions were, however, widely experienced; and cases of excessive rusting were isolated. It did not appear logical to accept them as sufficient to account for the troubles in question.

(3) *De-gaussing.*

This war-time measure involved the presence of a magnetic field which, it was suggested, might induce currents in the turbine shafting sufficient to set up galvanic corrosion. Calculation of the maximum potentials generated by the field in question showed that they would be of an order less than the ordinary contact potentials of the different metals, and incapable of producing any marked galvanic effect. Moreover, de-gaussing was universal, and heavy corrosion relatively infrequent.

(4) *Detergent effects of new oil supplies.*

The suggestion here was that oils of the naphthenic type sometimes supplied in U.S.A. and Canada, were likely to bring into circulation rust and oil sludges previously adhering harmlessly to casings, thus emulsifying an excessive quantity of water into the oil, which water would in turn cause fresh corrosion. The absence of records of oil supplied made it impossible to secure evidence in support of this explanation but the trouble-free experience of most ships known to have made the change of oil tended to discount it. Laboratory attempts to produce the sequence of effects described were unsuccessful.

Reasons for regarding salt intrusion as the most frequent cause of heavy corrosion.

(1) Although no more than a trace of salt is usually present in turbine lubrication systems, significant amounts were found in each case of severe corrosion investigated, and no case has so far been heard of where salt could not be found.

(2) Salt water is a more potent corrosive agent than distilled water, which is commonly present in turbine systems and does not cause corrosion on the scale investigated.

(3) A very close reproduction of the heavy corrosion investigated was twice obtained by adding sea-water to engines previously in good condition. (See pp. 673 to 679.

(4) The number of salt-water leakages into machinery increased during the war owing to underwater concussion. This coincided with an outbreak of corrosion trouble, not generally, but in a limited number of ships, nearly all being types making extensive use of depth charges.

PREVENTIVE MEASURES.

(1) *Corrosion-inhibiting oils.*

A very important contribution to the solution of corrosion problems is the development of corrosion-inhibiting oils. In a turbine system the oil necessarily has a great part in rust prevention—that is to say, the moisture ordinarily present would play havoc if it were not, in the main, held out of contact with the steel of the system by the much larger volume of oil. In general, water droplets in contact with working surfaces are mechanically displaced by oil while the turbine is running, but other surfaces on which water can rest as it settles out of the oil are liable to corrosion, the risk being greater the less the degree of turbulence. When the turbine is not running, the relative immunity enjoyed by working surfaces disappears, and "water-marks" on journals, etc. result from the presence

of any considerable proportion of water in the oil. The protective efficiency of an oil can be greatly increased by the addition of polar materials which form a molecular film on the steel, so hindering the contact between steel and water. Figs. 2 and 3, illustrate the effect of the polar material. Disks of nickel steel were submerged to a depth of 3/8 inch in (a) plain mineral oil SMLO; (b) corrosion-inhibiting oil 2190T. Water was added in droplets which sank through the oil and rested on the steel, as happens with machinery after shutting down. The photographs were taken after four days standing at room temperature. It will be seen that the corrosion-inhibiting oil 2190T prevented corrosion by distilled water whereas the plain mineral oil SMLO did not. This lasted for twenty days, after which all the droplets showed rust.

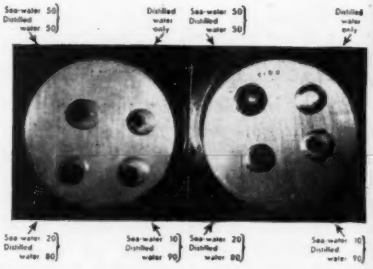


FIG. 2.—Plain mineral oil. FIG. 3.—Corrosion inhibiting oil 2190T.

Unfortunately the results are not nearly so good if salt be present in the water. The condition of the salt-water droplets shows that, although oil 2190T delays rust-formation, especially by water not too heavily contaminated with sea-water, it cannot prevent rusting even for the short space of four days. It was observed further that, while each droplet remained unattached to the steel and would move freely with movement of the oil so long as no rust was apparent, it became attached to the steel with the beginning of rusting and if forcibly disturbed would break up, leaving behind a wet rust-patch to which another droplet would readily adhere.

It may therefore be concluded that salt water causes disruption of the protective oil film far more readily than does fresh water and, once corrosion has begun, the re-establishment of a protective film of oil may be prevented, even after the turbine has been started up. Hence it is unsafe to rely on corrosion-inhibiting oils to ensure protection after contaminated water has entered the system. This is apparent from Case IV cited above. With pure distilled water, the protection afforded by corrosion-inhibiting oils is good but, even so, the liability of the protective film to break down if unrenewed, for example during a prolonged shut-down, should be taken into account and the amount of water kept to a minimum. The difficulty in providing more effective inhibitors lies, largely, in the fact that most suitable polar materials are also emulsifying agents, and on that account cannot be tolerated in so large a proportion as would otherwise be desirable. It is found also in the fact that when the supernatant oil is locally displaced by water the protective film (necessarily very thin) is deprived of its source of material for replacement. Thus the slightest discontinuity, however caused, enables corrosion to gain a foothold from which it can spread and undercut the neighboring film.

(2) *Water-soluble corrosion inhibitors.*

The possibility of dissolving a corrosion inhibitor in the water in the system has hitherto been neglected, doubtless on account of several obvious objections. There are not many water-soluble inhibitors which do not react either with lubricating oil or with one of the non-ferrous metals used in a forced lubrication system, and apart from this the wastage of inhibitor due to continual extraction of the water by

centrifugal separators could entail considerable expense and difficulty.

The war-time epidemic of heavy corrosion, however, provided an incentive to explore this line of attack in the assurance that, even if the drawbacks could not be eliminated altogether, they would need to be serious indeed to outweigh the consequences of inaction.

Although cases have been known in which sea-water was present almost undiluted in lubrication systems, there did not appear to be any need to employ the full strength of inhibitor solution necessary to combat this concentration of salt, since it is always possible to extract the bulk at least of the water from the system and by adding distilled water to dilute what remains. Indeed, after the ingress of sea-water has been stopped, the system can be cleared of salt by repeating this flushing process a sufficient number of times. It is during the interim period between the occurrence of a leak and the complete elimination of salt that the inhibitor is most needed, and it was considered that most attention should be given to the dosage necessary to counteract the effect of sea-water diluted with distilled water to one in four or weaker.

Laboratory Work.

General consideration of the possibilities pointed to sodium nitrite as the most promising inhibitor. It was known to be highly efficient in distilled water and with low concentration of salt, and there was no obvious reason to expect reactions with oil or non-ferrous constructional metals. It had, moreover, been used successfully to prevent rusting of steel by soluble oil emulsions.

As already remarked, rusting in forced lubrication systems is controlled to a large extent by the presence of oil but the effectiveness of the control depends on the various circumstances which determine how much of the steel shall be oil-wet and how much water-wet and for how long.

It was therefore decided to make two investigations (a) a purely metallurgical one, no oil being present, and (b) a simulation of practical conditions, with oil, air and aqueous solutions circulated over the metals.

The proportions of nitrite required to protect steel against different concentrations of sea-water in absence of oil could then be taken as adequate in practice, and if such proportions had no ill effect when used under simulated practical conditions it was considered that the way would be open for a full-scale trial.

The metallurgical work was arranged with Dr. Hoar of the Corrosion Section, Metallurgical Laboratories, Cambridge University, under an existing research contract with the Ministry of Supply, and the tests in presence of oil were made by the author.

(*1*) *Metallurgical Investigations.*

Dr. Hoar's work will be published in detail elsewhere. The principal conclusions reached were:—

(a) Any distilled - water / sea - water moisture (up to 50 per cent sea-water) can be rendered completely non-corrosive to steel at 25 deg. C. and at 60 deg. C. by the addition of a percentage of sodium nitrite equal to one-fifth of the sea-water percentage, except that for low sea-water concentrations at 60 deg. C. rather more nitrite in proportion is required.

(b) Smaller amounts of nitrite, not giving complete protection, convert general rusting into local pitting which, in the case of the machinery in question, is probably less dangerous than general rusting.

(c) The amounts of nitrite needed for steel protection have a decidedly beneficial influence on the corrosion of white metal, a slightly beneficial influence on the corrosion of copper, and no appreciable influence on the corrosion of brass.

(d) Sodium nitrite is effective only in neutral or alkaline solution. The *p*H value of the solution should preferably be about 8 and should not fall below 7.

Dr. Hoar's results for mild steel are given in further detail in Fig. 4.

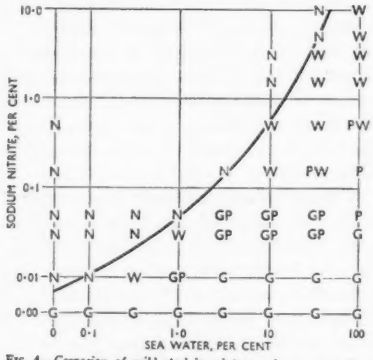


FIG. 4.—*Corrosion of mild steel in mixtures of sea water, distilled water and sodium nitrite*

N = no corrosion
W = water-line pitting
P = pitting over the whole surface
G = general attack over most of the surface

A comprehensive series of tests was later carried out by Dr. D. Wyllie of the Chemical Department H. M. Dockyard, Portsmouth. In this the effect of varying proportions of sea-water on the corrosion of steel is studied, and it is shown that the percentage of sodium nitrite required to inhibit corrosion can be reduced by adding quite small amounts of other substances. These results also will be published elsewhere.

(*2*) *Simulation of Working Conditions.*

For the tests under simulated practical conditions the author used a cast-iron gear pump fitted with steel gears and copper and brass connecting tubes, arranged to circulate oil and water over specimens of white metal, copper, aluminium bronze, aluminium and mild steel sheet. The charge of oil and water was circulated ten times per minute and air was admitted to the suction side of the pump to maintain the liquid in a thoroughly aerated condition.

It was found that, with sufficient heat applied to the delivery pipe to maintain the temperature in the reservoir at 80-85 deg. C. Admiralty Special Mineral Lubricating Oil (SMLO) and distilled water rapidly formed a dense emulsion, and there was considerable frothing. The water evaporated quickly despite the addition of a vent tube as a reflux condenser to the reservoir, and within twelve hours the oil became noticeably oxidized. It appeared, therefore, that the conditions were severe although the temperature of the bulk of the oil was much lower than the temperatures ordinarily used in laboratory oxidation tests. Probably the intense aeration and the presence of water and metal dust from the pump contributed to the acceleration of oxidation, but the local rather than general application of heat to the system is likely to have been the principal factor. This local heating, however, is in conformity with conditions in practice, where the shearing of the oil film generates a large amount of heat in a very small volume of oil.

The apparatus was next run at the lower reservoir temperature of 50 deg. C. with SMLO and 18 per cent of distilled water. In twelve hours there was little change in the oil and no corrosion of the metals, though an emulsion which did not completely separate for some hours was formed. In view of these results, a temperature of about 70 deg. C. was thought suitable for further tests.

The three twelve-hour tests recorded in Table 1 gave no indication of inter-

Table 1. Effect on metals of (a) distilled water, (b) salt water, (c) salt water plus sodium nitrite mixed with oil, in a twelve-hour circulation test at 70 deg. C.

Circulating mixture	Corrosion of metal specimens			
	Steel	Copper and aluminium bronze	White metal	Aluminium
180 ml. oil (SMLO) 40 ml. distilled water	None	Slight stain	None	None
180 ml. oil (SMLO) 30 ml. distilled water 10 ml. artificial sea water	Considerably pitted	Slight stain	None	None
180 ml. oil (SMLO) 30 ml. distilled water 10 ml. artificial sea water 2·8 gm. sodium nitrite (=7 per cent of the water)	None	Slight stain	None	Slight whitening

action between sodium nitrite and lubricating oil, but it was found that the nitrite emulsion took somewhat longer to separate. Copper and white metal were practically unaffected but aluminium showed definite signs of attack. Steel was fully protected, although considerable rusting occurred in presence of salt water without nitrite.

A further test was made over the longer period of 34 hours (see Table 2) and with full-strength sea water, using 7 per cent and 14 per cent of sodium nitrite. The latter high concentration was adopted to show up any unwanted effects as clearly as possible, rather than with the expectation of its being necessary in practice.

Table 2. Effect on metals of undiluted sea water plus (a) 7 per cent sodium nitrite (b) 14 per cent sodium nitrite in a thirty-four hour circulation test at 65 deg. C.

Mixture	Corrosion of metal specimens			
	Steel	Copper and gun-metal	White metal	Aluminium
180 ml. oil (SMLO) 40 ml. artificial sea water 2·8 gm. sodium nitrite (=7 per cent of the water)	Slight local pitting	Slight stain	Dulled and very slightly pitted	25 per cent of area corroded
180 ml. oil (SMLO) 40 ml. artificial sea water 5·6 gm. sodium nitrite (=14 per cent of the water)	None	Slight stain	Slightly dulled	80 per cent of area corroded

Again there was no indication of harmful interaction between nitrite and oil, and the corrosion effects were of the order to be expected from Dr. Hoar's results. Aluminium was included in these experiments in order to gain a rough idea of the intensity of attack to be expected. Normally in steam-turbine practice it is either not used at all, or is employed only in the form of robust castings such as gear-box covers of aluminium silicon. It was concluded that these would not suffer any harm unless the exposure were far more severe and prolonged than that contemplated.

(3) Effect of Sodium Nitrite on Demulsification of Oil.

The slowing down of separation of emulsified oil and water when nitrite was present was noted repeatedly, and pointed to a possible difficulty in removing water by means of the centrifugal separator, especially from old, partially oxidized oil. Since the addition of nitrite is proposed only as an emergency measure to prevent serious damage to machinery, the possibility that a charge of emulsified oil might have to be set aside for special separation treatment, or discarded altogether, could doubtless be accepted. A few experiments were made, however, in order to assess the extent of the difficulty

A partially oxidized charge of SMLO (I.P. demulsification value = 470) mixed with undiluted sea water containing 14 per cent of sodium nitrite, was taken as representing thoroughly adverse conditions and the rate of breaking of the emulsion formed from this mixture was compared with the rate for the same oil when mixed (a) with distilled water and (b) with sea-water alone. Each oil-water mixture (200 ml.-80 ml.) was circulated in the pump apparatus for fifteen minutes, producing a close emulsion. This was quickly transferred to a graduated tube as used in the Institute of Petroleum test for demulsification, immersed in a water-bath at 200-203 deg. F., and the rate of decrease of the emulsion layer (as oil and water settled out) observed. The rates for the three mixtures are given by Fig. 5.

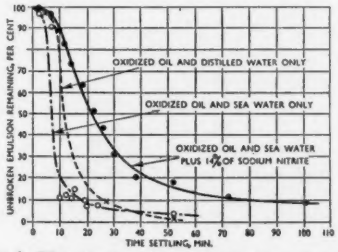

UNBROKEN EMULSION REMAINING, PER CENT
TIME SETTLING, MIN.
OXIDIZED OIL AND DISTILLED WATER ONLY
OXIDIZED OIL AND SEA WATER ONLY
OXIDIZED OIL AND SEA WATER PLUS 14% OF SODIUM NITRITE

FIG. 5.—*Effect of sodium nitrite on the break-up of emulsions during settling at 200-203 deg. F.*

It will be seen that, while there is no very marked difference between the curves for distilled water and sea-water the time required for the sea-water

containing nitrite to separate out to any given extent is longer. In fact, during the sea trials to be described later, no difficulty arose.

(*4*) *Tests for Salt and for Sodium Nitrite.*

It was thought desirable, before initiating practical trials, to have convenient methods of chemical control available for use in a ship.

The method of titration by silver nitrate used for determining salinity of boiler feed water is quite satisfactory when applied to water derived from a turbine oil system, the presence of traces of oil causing no difficulty. Similarly the approximate *p*H value of the water is ascertainable in a few moments by means of a universal indicator such as BDH 678.

If the oil is reasonably clean, its water content may be estimated by settling. The tube (Fig. 7) is filled with white spirit to B. A little sulphonated castor oil or similar "wetting agent", diluted half and half with water, may be added to assist separation of water and its volume at A noted after settling.

A representative sample of oil is drawn, preferably from a test cock while steaming and, without giving time for water in it to settle, 100 ml. is poured into the tube, reaching to mark C. It is then mixed with the white spirit by inverting and shaking. After completely settling in a warm place, droplets having been dislodged from the sides of the tube by spinning it about its axis and sharply reversing direction, the volume of the watery layer is again noted. Each ml. increase represents one per cent of water in the oil.

Methods for estimating water in dirty oil need not be described here.

Since an over-dose of sodium nitrite will do no harm it is seldom necessary to determine its concentration analytically. A method suitable for engine-room use was, however, worked out and may be of interest. It proved satisfactory during sea trials. The method is based on the reaction of nitrite with sulphamic acid, suggested by the Chemical Inspection Department, Ministry of Supply:—

$$NaNO_2 + NH_2HSO_3 = NaHSO_4 + N_2 + H_2O.$$

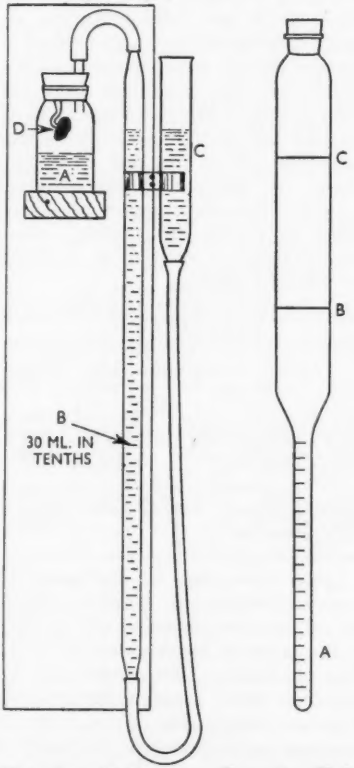


FIG. 6.—*Apparatus for determining the sodium nitrite concentration.*

FIG. 7.—*Tube for estimation for water content.*

The determination is carried out as follows (Fig. 6):—

Exactly 1.00 ml. of the water is delivered from a 1 ml. pipette into the bottle A, which is half-filled with distilled water. The stopper is inserted, and water poured into C (the water levels should remain steady, indicating absence of air-leaks. With the stopper removed, and C adjusted to bring the

water-level in B near to the zero mark, a filter-paper packet containing about 0.25 gm. of sulphamic acid is attached to hook D and the stopper firmly replaced. The level in B having been noted, A is swirled to detach the packet and open it up in the water.

Evolution of nitrogen gas will occur, and C is readjusted to equalize the water levels in B and C. The volume of nitrogen is then read off from the graduations on B.

Since 1 gm. of sodium nitrite yields 350 ml. of nitrogen gas at 67 deg. F. and normal pressure, the percentage present in the sample is

$$\frac{\text{Volume of nitrogen evolved (ml.)}}{3.5}$$

Results are generally correct within a tenth of the percentage of nitrite present.

TRIALS IN SHIPS' MAIN ENGINES.

(1) Use of Sodium Nitrite to prevent Corrosion by Sea Water.

Through the interest taken in the author's proposals by the Engineer-in-Chief's Department at the Admiralty it became possible to carry out practical trials.

Full-scale tests were begun cautiously with the daily introduction of small quantities of sodium nitrite into the forced lubrication system of a destroyer's turbines, the quantities being adjusted to 0.2 per cent of the water entering from the steam glands, which amounted to 117 gallons during 160 hours continuous steaming. No salt was present. At the end of the trip the journals showed no perceptible change. Old markings due to corrosion before the last refit were visible; otherwise their condition was normal.

In order to secure a positive result, permission was next obtained to put sea-water into both engines of an old destroyer in order to create corrosive conditions, and to add sodium nitrite to one engine only.

The opportunity was found in a twelve-year old destroyer already

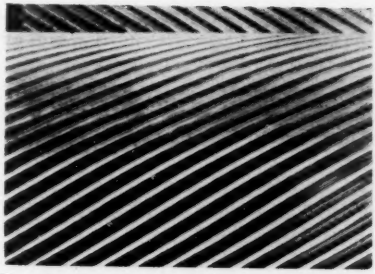


FIG. 8.—Port engine pinion at start of experiment.

marked for breaking up, but based for a few months on Portland for occasional anti-submarine exercises in the Channel. The intermittent use of her engines provided excellent conditions for a corrosion test, and the fact that the ship was never far from her base greatly reduced the risk of complications in the event of an unexpected breakdown.

Her previous record showed no trouble with the forced lubrication system, and in a preliminary examination the journals and gearing were found to be in good condition, smooth to touch but lightly pitted, with no signs of active corrosion. There was no noticeable difference in the condition of the port and starboard engines. This is illustrated by Figs. 8 and 9.

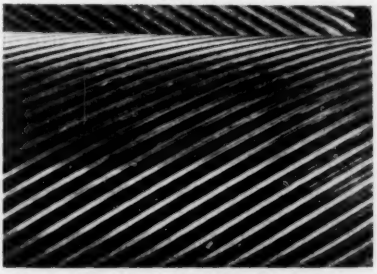


FIG. 9.—Starboard engine pinion at start of experiment (no difference).

The old lubricating oil, which had so far circulated through both systems with the interconnecting valves open, was removed, and the drain tanks

cleaned by hand. The port and starboard systems were then isolated by closing the cross-connections and each drain tank charged with 180 gallons of new SMLO, plus 30 gallons of distilled water and 10 gallons of synthetic seawater. The water present thus had a test salinity of 320 grains of chlorine per gallon. In the case of the port system only, 28 lb. (equivalent to 7 per cent), of sodium nitrite was dissolved in the water.

The ship then continued her duties with the salt solution in the starboard forced lubrication system and the salt solution plus sodium nitrite in the port forced lubrication system. In accordance with normal practice the forced lubrication pumps were run for two hours on each occasion before starting main engines, and for half an hour after stopping.

In the early stages of the trial an attempt was made to estimate the quantity of condensed steam entering the system by taking float gauge readings on the drain tanks when in harbor. These readings proved erratic and unreliable, so recourse was had to sampling the mixed oil plus water from a bearing test cock while under way, and determining the percentage of water present by simple settling tests. The salinity and the sodium nitrite content of the water were also determined.

It was found that only 10 per cent of water was present instead of the 18 per cent expected, the balance having disappeared, possibly into pockets in the system. It was decided to operate the trials with this proportion rather than make any further addition. The salinity and nitrite content of samples of water from the two systems were also slightly below the concentrations intended, probably owing to the presence of condensed steam in the system at the start. As the trial proceeded, however, it was found that very little condensed steam was entering and that it would be unnecessary to extract water until the end of the run. In these circumstances fresh additions of seawater or nitrite were not made, although the concentrations show a small progressive decrease with lapse of time

Table 3. Progressive decrease in concentrations of salt and sodium nitrite in the lapse of time

Hours steamed	Sodium chloride		Sodium nitrite		pH value	
	Starboard (without nitrite), per cent	Port (with nitrite), per cent	Starboard (without nitrite), per cent	Port (with nitrite), per cent	Starboard (without nitrite)	Port (with nitrite)
50 hours	0·6	0·7	0·2	6·0	8·8	8·8
116 hours	0·6	0·7	0·2	5·0	8·8	8·8
158 hours (end of trial)	0·4	0·6	0·3	4·0	8·8	8·8

(Table 3). A small percentage of sodium nitrite found its way from the port to the starboard system.

A first inspection of the main gear wheels was carried out fourteen days after the start of the trial. The ship had made nine trips totaling fifty hours steaming during that time. The port wheel showed no change in appearance, but the starboard wheel was unmistakeably rusty.

Thirty days later the Engineer Officer of the ship reported that the starboard "Autokleen" oil strainer had become choked with rust, the self-cleaning mechanism being found immovable. A new element was fitted but this seized up soon afterwards and the strainer had to be by-passed.

A second inspection of the gears was made on the forty-ninth day from the start, the ship having been at sea on twenty-one days and completed 110 hours steaming. There was still no change in the nitrite-protected port engine, the journals being in excellent condition except for old water marks, and the gearing showing no signs of new corrosion. The starboard journals, however, were pitted with fresh rust despite the self-cleaning effect of rubbing surfaces, and the pinions showed new corrosion. The main gear wheel was so covered with red rust that very little bright steel could be seen.

The thirtieth and last trip was made on the seventy-second day, bringing the total hours steamed to 158. A week later a final inspection was carried out.

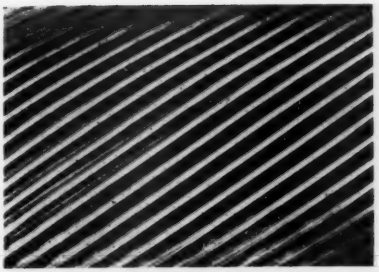


FIG. 10.—Protected port engine surfaces smooth and bright.

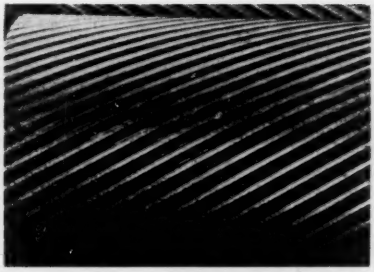


FIG. 11.—Unprotected starboard engine surfaces rusty.

The port main gear wheel remained bright and free from fresh rust with the exception of a few spots probably caused by condensed moisture dripping from the casing during the seven days prior to inspection. The port pinions and journals still showed no change in condition since the start of the trial. Various photographs to bear out this conclusion were taken, two of which are reproduced in Fig. 10 and Fig. 12. No measurable weardown of the bearings had taken place during the trial. The main thrust collar was in good condition, as will be seen from Fig. 14. The rusting visible on the outside edge was not active or of a serious nature and had probably taken place over a number of years. It was agreed by all concerned that the salt water had been prevented from having any adverse effect and the engine was in normal condition. It could be turned freely by the hand turning gear.

The starboard engine, on the other hand, was not far from breakdown. The main gear wheel was very badly corroded and covered with active red rust. The pinions were likewise badly corroded, though not quite to the same extent as the main wheel. The journals showed an average wear-down of 0.026 inch during the trial against approximately 0.003 inch during the whole previous life of the engine. Journal corrosion was very bad, and the white metal had started to pick up. The engine could not be turned by hand. It was found that during standing the suction line to the starboard forced lubrication pump had become choked with rust, and both pump and pipe had to be dismantled and cleaned. It was clear that the engine could not have run much longer Fig. 11, and Figs. 13 and 15, illustrate its condition and should be compared with the corresponding views of the port engine in Fig. 10, and Figs. 12 and 14, keeping in mind that the engines were alike at the start of the trial, and that photographs in black and white, though clear enough when closely examined, cannot reproduce the strong contrast between the two engines which was immediately apparent on inspection.

The non-ferrous metals in each engine were free from marked corrosion with the exception of the aluminium silicon gearbox covers. These showed signs of attack, though not to a damaging extent.

It was agreed that this trial had shown the use of sodium nitrite to be satisfactory for the purpose in view.

(2) Use of Sodium Nitrite to Arrest Existing Corrosion.

Experience in other ships having shown that salt-water corrosion, once started, is very persistent even after removal of the salt water from the oil-circulation system, it was considered to be of interest to discover whether the corrosion of the starboard engine could be arrested by circulating sodium nitrite, and since the ship had to re-

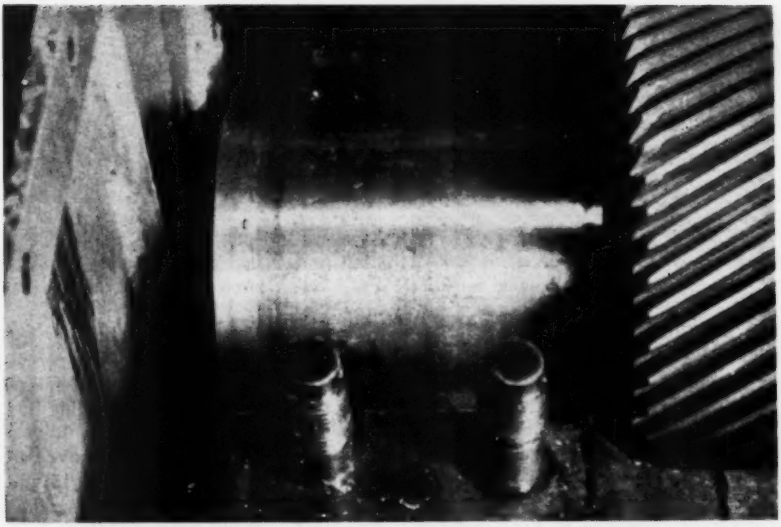


FIG. 12.—Protected port engine journal with no wear down and surface smooth and bright apart from old water marks.

main in service for a time the opportunity was taken to make this further experiment.

The starboard forced lubrication system was emptied and cleaned by circulating two proprietory cleaning fluids

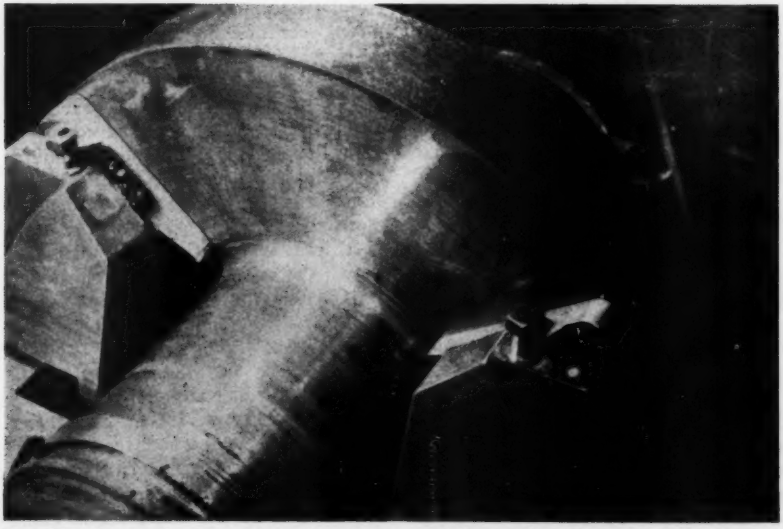


FIG. 14.—Protected port engine after three months' operation with salt water present in the lubricating system.

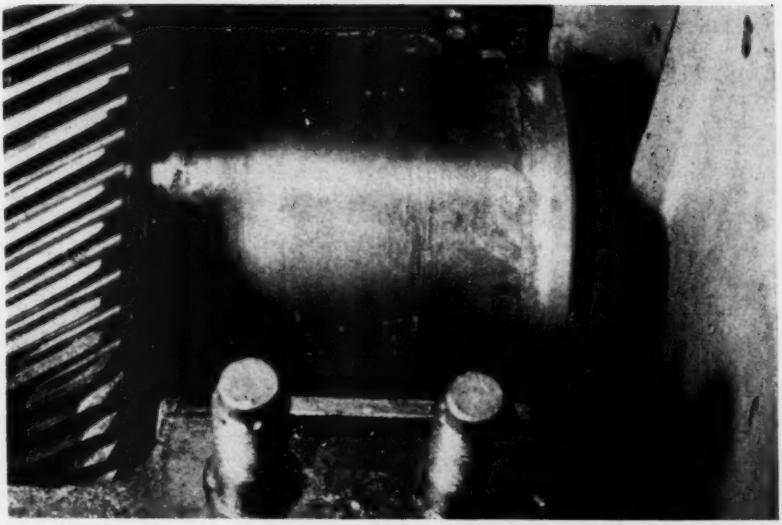


FIG. 13.—Unprotected starboard engine with 0.026-inch wear down and old water marks obliterated by new corrosion.

in succession, flushing with water after each. This removed most of the loose rust. The system was then re-charged with new oil (SMLO) plus thirty gallons of a 2 per cent solution of sodium nitrite, and the ship returned to sea.

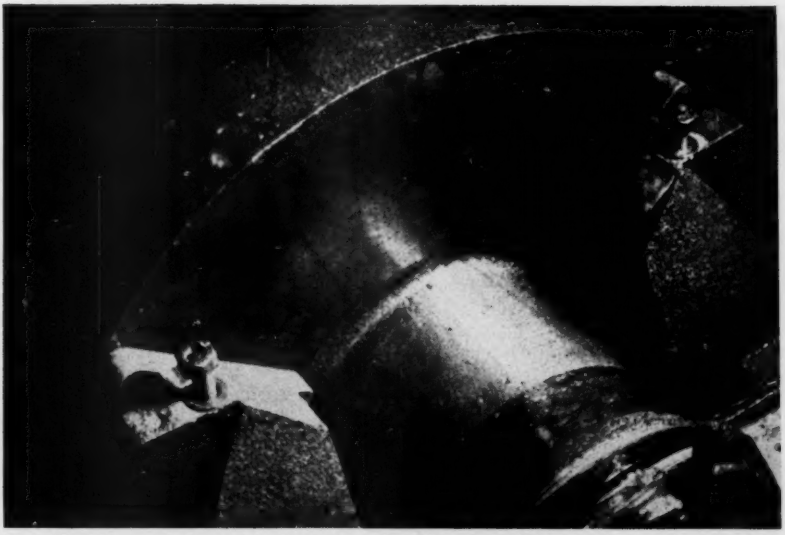


FIG. 15.—Unprotected starboard engine after three months' operation with salt water present in the lubricating system.

After four days at sea, this solution, containing more rust detached by the movement of the machinery, was discharged by centrifuging and replaced by a second thirty gallons.

After fourteen days there was no sign of active corrosion, the rubbing surfaces of the gear teeth being bright and the rest covered with old hard rust of dark appearance. The sodium nitrite solution was next removed by the separator and replaced by thirty gallons of distilled water. The ship continued in service for three weeks, and final examination prior to her departure for breaking up showed the condtion of the gears to be unchanged, with no indication of corrosion having recommenced. The journals, though covered with non-active rust, had become comparatively smooth, and wear-down over the period dating from the introduction of nitrite was 0.003 inch, as compared with 0.026 inch for the previous trial. This experiment gave a clear indication that if the mechanical condition (degree of wear and tear) of the machinery permits, a system which has been severely corroded may be cleaned up and rendered fit for further service by the treatment described, without the expense and delay of mechanical overhaul, and that corrosion which has been checked by the use of sodium nitrite will not recommence in the presence of fresh water. Further confirmation has since been obtained from a practical application of the treatment (p. 681).

(3) *Use of Corrosion-inhibiting Oil.*

While the experiment just described was being carried out in the starboard engine, the port engine was used to confirm the conclusion (already arrived at from laboratory work and from Case IV p. 666), that a corrosion-inhibiting turbine oil to meet U.S. Navy Symbol 2190T does not ensure immunity from corrosion by salt water. The port forced lubrication system was cleaned in the same way as the starboard, but was recharged with 180 gallons of oil 2190T. To remove any residual solids the separator was kept in action for several days. 7½ gallons of synthetic sea-water and 22½ gallons of distilled water were then added, and the ship proceeded on her normal duties. After fourteen days the gears which of course were bright and practically corrosion-free at the outset, were found to be speckled with patches of dark, hard rust, and with areas of active corrosion. The salt water was removed by the separator, and 40 gallons of distilled water containing 7 per cent of sodium nitrite were then added to arrest the action. A final examination three weeks later revealed no active corrosion. Weardown of journals was 0.007 inch. This experiment therefore confirmed that oil 2190T cannot be relied upon for protection against sea-water, when present to the extent of one part in four of the total amount of water, and again showed that existing corrosion can be arrested by the use of sodium nitrite.

Recommendations.

(1) *Routine Precautions against Sea Water Corrosion.*

It has already been mentioned that the salinity test used for boiler feed water can be applied to the slightly oily water extracted from a turbine oil system. In practice, it is important to carry out this test at the time when the amount of salt water, if any, is at a maximum. Entrance of sea-water is most likely to occur when there is no oil pressure on the system, and the test should therefore be made on water separated when in harbor as well as at sea. The possibility of intake of bilge water on the suction side of the lubricating oil pumps, on the other hand, renders it desirable to test at intervals of, say, one week in all circumstances. Naval practice is to test once a week in harbor and daily at sea.

(2) *Action in Cases of Slight Contamination by Salt Water.*

With plain lubricating oils in use the

tendency to cause corrosion increases with the concentration of salt from zero upwards, and it is advisable to resort to flushing operations if ever the concentration exceeds 10 grains chlorine per gallon. (The salinity of sea-water is about 1,400 grains per gallon). Flushing is carried out by repeatedly introducing distilled water, circulating it with the oil while the engines are running, and removing it by centrifugal separation. Provided the amount of water is not excessive and it is mixed with the oil by running the circulating pumps before starting the engines, or by adding it slowly while steaming, there need be no fear of damage by failure of lubrication. Mixtures of more than one gallon of water to nine of oil have been used for long periods without any harm, though so large a quantity should not be present if the engines are to be run at full power. Up to 3 per cent is considered quite safe.

(3) *Action in Cases of Heavier Contamination by Salt Water.*

It is taken for granted that when a serious amount of sea-water is detected, prompt steps will be taken to locate and remedy the leakage. In such circumstances, when the salinity exceeds, say, 30 grains chlorine per gallon, the danger of corrosion will be considerable, and flushing with sodium nitrite solution should be employed. A solution containing 7 per cent of nitrite is suitable and a stronger solution should be unnecessary, since the salinity, if excessively high, can be brought to a moderate figure by the first flushing operation. When repeated extractions and replenishments of nitrite solution have sufficiently reduced the salinity of the water, a few days steaming should elapse to give the nitrite time to arrest all corrosion before the last charge of solution is finally flushed away with distilled water and normal operation resumed.

Salvage of Machinery in a State of Advanced Corrosion.

When the presence of salt water has remained undetected and repair work is necessary, it is important to prevent further corrosion during and after the refit. The best method is that recommended above, since circulation of the nitrite ensures that every center of corrosion will be reached. However, it is not always possible to run the machinery, and some alternative is then necessary.

In Case IV, described on p. 666 various unsuccessful efforts were made to save the corroded gearing by cleaning away the rust, and further experiments were carried out with one of the condemned gear wheels after it had been put ashore. A number of large heavily rusted bolts were taken from the flange, and used for small-scale tests. When one of these was simply filed bright, hundreds of corrosion pits remained dotted about the surface and within a few hours an eruption of moist black oxide formed over each pit, slowly turning red and spreading until within a week the surface was again completely rusty.

Immersion in boiling water extracted salt, but only slowly. Fig. 16 shows that water-washing is not effective quickly enough to be useful when rusty machinery is being cleaned by hand.

Scrubbing with a solution of sodium nitrite, and coating with rust-inhibiting oil or water-displacing fluid were likewise ineffective, clearly because the sub-surface centers of activity, full of salt water and corrosion products, were not reached by such means.

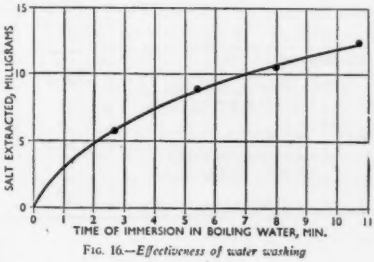


Fig. 16.—*Effectiveness of water washing*

Table 4. Arrest of corrosion showing effect of various treatments applied to heavily rusted bolts from gear wheels (see Case IV above) after degreasing and filing bright.

Treatment	Standing period	Condition	Further standing period	Condition
None	6 hours	Beads of wet rust over the corrosion-pits	7 days	Surface 100 per cent rusty
Washed several times with hot water	1 day	Rust formed over most pits but in smaller volume	7 days	Surface 70 per cent rusty
Plastered with nitrite paste; wiped off after four days	1 day	Rust formed over a few pits only	16 days	Surface 30 per cent rusty
Washed with hot water and plastered with nitrite paste; wiped off after eight days	1 day	No rust	20 days	No rust
Plastered with nitrite paste (about 1/20th inch thick)	7 months in damp atmosphere with occasional slight condensation of moisture	On wiping the surface clean, no rust was apparent	—	—

It seemed likely that if a solution of solium nitrite could be made to penetrate these centers it would provide the remedy, and that the requirement was to keep such a solution in contact with the rusty surfaces for as long as might be necessary for it to diffuse into the pits and crevices. To this end a strong solution was made into a paste. The effect of the paste is shown by tests recorded in Table 4. Each bolt was degreased and filed bright before treatment.

Meanwhile similar trials were put in hand on areas of the journal and toothed rim of the wheel, which stood covered with sacking in a weather proof shed. The surfaces were first

Table 5. Effect of various treatments applied to parts of heavily rusted main gear wheels (see Case IV above) after degreasing and cleaning with carborundum cloth.

Test Ref.	Rust-arresting treatment	Standing period	Condition	Subsequent treatment	Standing period	Condition
On wheel teeth						
A.1	Coated with plain mineral lubricating oil	1 month	Slight rusting	Wiped clean	7 months	About 80 per cent of area rusted
A.2	Coated with corrosion-inhibiting oil	1 month	Slight rusting	Wiped clean	7 months	About 40 per cent of area rusted
A.3	Coated with nitrite paste	1 month	No rust	Wiped clean	7 months	About 15 per cent of area rusted
On journal surface						
B.1	None	—	—	—	7 months	Rusty
B.2	Coated with plain mineral lubricating oil	—	—	—	7 months	Rusty, but slightly less so than B.1
B.3	Coated with nitrite paste	1 month	No rust	Paste wiped off	7 months	About 10 per cent of area rusted
B.4	Coated with nitrite paste	1 month	No rust	Paste wiped off and plain mineral lubricating oil applied	7 months	No rust
On wheel teeth						
C.1	Scrubbed with 20 per cent sodium nitrite solution and coated with nitrite paste	3 weeks	No rust	Paste wiped off	7 months	No rust except small patches on tops of teeth about 1 per cent of total area
C.2	Scrubbed with 20 per cent sodium nitrite solution and coated with nitrite paste	3 weeks	No rust	Paste wiped off and plain mineral lubricating oil applied	7 months	No rust

freed from grease with rag soaked in trichlorethylene and rubbed bright with carborundum cloth. The results were as shown in Table 5.

It will be seen that oils fail to stop rusting (A.1, A.2, B.2) though the corrosion inhibiting oil had more retarding effect than the plain mineral oil. Nitrite paste stopped rusting completely (A.3, B.3, C.1) and when oil was subsequently applied no rust formed for a further seven months at least (B.4, C.2). The corrosion-inhibiting effect of the paste continued after it was wiped off, and without subsequent oiling, to the extent that very little rust formed within seven months (C.1).

Another experiment, designed to test the treatment under extreme conditions, was carried out on the cover of a water softening unit which had been in use for fifteen years. During this time nothing had been done to counteract the effect of continual spilling of salt (used in the softener) over the cover, and excessive rusting had resulted. By scraping and scratch-brushing some 14 oz. of rust scale was first removed. The surface, approximately one square foot in area, was next thoroughly scrubbed with water and allowed to dry. One half was coated with nitrite paste and left for thirty-eight days, the paste being renewed once during that time. The whole surface was then again scrubbed with water, allowed to dry, and painted with cellulose varnish pigmented with aluminium powder. Within eight days the portion not treated with nitrite paste became dotted with beads of moist red rust erupting through the paint. These increased until fifty days later the cover presented the appearance shown in Fig. 17, one half rusty, the other with paint intact. It was then once more scrubbed with water and re-painted all over, but within fifteen days rust was again breaking through the paint on the untreated section thus showing the persistent character of deep-seated salt corrosion.

FIG. 17.—Partially treated water softener cover, 14 ozs. of rust were removed before the experiment was carried out.

From these experiments it was concluded that nitrite paste could be employed to stop the progress of corrosion during a refit.

EXPERIENCE WITH SODIUM NITRATE TREATMENT IN THE ROYAL NAVY.

Since the completion of the work described above, the nitrite treatment has been used in two ships.

In September 1947 the turbine oil system of a minesweeper was found to contain a quantity of almost pure seawater, which has caused extensive rusting of gears and journals. The entry of sea-water was traced to split tubes in an oil cooler.

The method adopted was as follows:—

(1) The surfaces of journals and gear teeth were cleaned with white spirit and rag, surface rust rubbed off with emery cloth, and the whole scrubbed with 5 per cent sodium nitrite solution. Nitrite paste was applied as a temporary protective to the journals until these operations were completed.

(2) The paste was removed and 2½ gallons of a 3 per cent solution of sodium nitrite added to each hundred gallons of oil in circulation. The ship then proceeded to sea.

(3) After six hours steaming the solution was withdrawn by means of the centrifugal separator.

(4) More nitrite solution was added and the procedure repeated until the salinity of the solution discharged was less than 10 grains of chlorine per gallon.

(5) Processes (2) to (4) were repeated using distilled water in place of nitrite solution.

(b) The oil system was emptied and re-charged with new oil.

Beyond the re-metaling of one or two scuffed bearings and the replacement of split oil-cooler tubes no repair action was necessary.

The ship returned to duty after this treatment, which had involved 2½ days steaming.

On examination four months later no sign of active corrosion could be found. The working surfaces were smooth and bright and no lubrication trouble had been experienced. A major refit was thus obviated.

During the autumn cruise of the Home Fleet a considerable amount of salt water was found in the forced lubrication system of a destroyer's engines soon after leaving Bermuda. The recommendations on p. 678 were followed (they now form part of a Fleet Order) and on the ship's arrival in home waters fourteen days later not only was the system free from salt, but there was no sign of any corrosion or damage. Without these precautions damage would undoubtedly have occurred.

INDUSTRY DISPERSAL FOR SECURITY.

ACKNOWLEDGMENT.

This article, which deals with a subject of considerable contemporary interest, is reprinted from "Signals" for March-April, 1949.

There is no known military defense against the atomic bomb itself except space. There is and will continue to be defense against any carrier which a potential enemy might use to deliver such a bomb. This defense alone is not sufficient.

The constantly increasing range of aircraft, together with the enormous destructive capacity of atomic weapons, makes it reasonable to assume that within the foreseeable future no area in the United States will be immune from possible attack because of its location alone. This assumption, coupled with the knowledge that the destruction or immobilization of a nation's vital industry will destroy its capacity to defend itself, makes it reasonable to assume that highly concentrated areas of vital industry and population will be the most attractive targets.

The risk of a sneak of unconventional attack becomes great because of the formidable and possibly decisive advantage which could accrue to a powerful enemy in modern warfare by choosing the time, place, and mode of attack. An enemy powerful enough to attack us might conceivably attempt a surprise attack having destructive effects many times those of Pearl Harbor. Sabotage and other methods of destruction would probably be attempted in an effort to prevent effective industrial mobilization for retaliation. Atomic bombs could be delivered simultaneously by plane against strategic industrial targets and by ship against our vulnerable ports.

Thus, it can be seen how location of industry assumes major proportions strategically. It further becomes evident that time is of the essence in establishing at least a pattern of location which will be strategically sound and which can be consistently followed.

Modern strategy presents these problems; Can industry, through location or relocation, achieve a reasonable degree of security if faced with the reality of the risks outlined above? Is it feasible? The answer to both of these questions is that dispersion of industry will go a long way toward combating a potential enemy's effort to cripple our industrial capacity by any mode of attack, conventional or otherwise. The security provided by industrial dispersion becomes more apparent when we measure the significance of modern strategy in terms of its possible application by a potential enemy.

What Can an Enemy Do.

Any aggressor nation planning to use atomic weapons will be faced with the problem of the number of atomic weapons that he could make available. This is an important basis for measuring an enemy's capacity for destruction. Assuming an enemy has, and can deliver, the atomic bomb, there remains another basic problem of how many he can make available. Atomic bombs are expensive. The essential fissionable elements—uranium and plutonium—are scarce materials. Any aggressor nation must, therefore, assess a target in terms of the amount of fissionable material it is considered profitable to allocate to its destruction. This means each target must be assessed in terms of its essentiality to the aggressed nation's war-making capacity.

There is the additional factor of the expense of bomb delivery. It will probably continue to be practically axiomatic that as range increases, relative payload decreases, and costliness in proportion to damage becomes greater. To this must be added whatever loss may be incurred in delivery through active interception

Thus, although modern strategy is based upon the fact that any area of this country can be attacked, the staggering cost in material and energy of such an extensive attack would require that it be planned so as to accomplish the maximum damage to our production facilities.

Factors in Relocation.

With this background, it is possible to develop a few general rules in considering strategic industrial location or relocation. The scarcity of the essential materials for the manufacture of an atomic bomb makes production so costly that we may reasonably assume that no country in the foreseeable future will ever have enough to afford to use one on each city of as few as 50,000 people, or on a congested industrial area of less than five square miles. It is, therefore, strategically desirable to plan industrial expansion so that further urban concentrations of more than 50,000 people may be avoided.

Atomic bombs exploded to date have destroyed almost everything within a one-half-mile radius of the zero point. Beyond the periphery of this first area, and extending to a distance for about one and one-half miles, they caused moderate damage to all structures. Allowing for future developments, it is not expected that an improved bomb would cause heavy damage beyond a distance of three miles from point of detonation. The possibility of a super bomb that will wipe out an entire state is too remote to warrant serious consideration.

Before evaluating present plant locations or proposed plans for relocation or expansion in terms of relative security or vulnerability in the event of attack, let us review and enumerate those implications of modern strategy and knowledge of modern weapons

Entrance to German factory found about two miles underground near Thil, France. Defense planners claim underground installations are not practical.

which make plant location of strategic importance, and establish the framework for our thinking in connection with this problem:

(1) Any attack upon the United States will be designed to cripple vital industry as a means of destroying our capacity for successful defense;

(2) Any such attack may be expected to be made suddenly and to be of such magnitude that general destruction in the areas affected can be assumed;

(3) The only lead time we can depend on is that required by a potential enemy to produce and be able to deliver atomic bombs, plus whatever time may be obtained by diplomacy;

(4) The present geographical location of any facility in the United States cannot be considered safe simply because of its remoteness from a potential enemy (although there are, of course, degrees of vulnerability);

(5) A high concentration of industry or population in a given area constitutes the best target for atomic or other modern weapons;

(6) Attack upon areas of industry concentration less than five square miles or urban concentrations of less than 50,000 people will probably not be economically feasible unless they contain specific installation of decisive importance to the nation's capacity for defense;

(7) A location less than three miles from a potential target center of an atomic bomb is seriously vulnerable.

Evaluation of Present Location.

The strategic planner of a potential enemy, in appraising the area in which your facilities are located, will be looking for certain objectives. It may be that your facilities are not the particular target he is seeking, but if there are others in the area that he has earmarked for destruction, the net results for you are the same. Here is a list of the kinds of facilities in which he would be most likely interested in a given area:

(1) An individual plant, producing a large percentage of a highly critical item, which, if destroyed, would have immediate far-reaching adverse effects on the production of that and other related items;

(2) All major developments, industrial or otherwise, which could be considered highly inflammable or explosive to the extent that their general destruction would substantially endanger other vital resources in the area. This would include such things as oil refineries, oil storage facilities, paint and chemical plants;

(3) Public utilities such as power plants and water systems which service a general industrial area producing important quantities of essential war materials;

(4) Railway, water, and other transportation facilities serving the area;

(5) Key establishments of the Arnold Forces;

(6) Dams and related works and bridges;

(7) Major air bases and air supply centers;

(8) Several plants all engaged in the production of related items of military importance or materials, such as trucks and steel.

Some Yardsticks.

It is suggested that the problem of assessing the vulnerability of your facilities be approached in the following manner:

Draw a *three-mile radius circle* about the periphery of your present facilities. Then, in the light of the possible objectives just outlined, list everything within this circle which you believe would be of interest to the strategic

planner of a potential enemy. It may not be possible to interpret exactly the kind of items which other plants within the encircled area would make during wartime, and the relative strategic value of such items to the war effort. However, with the general information available to you, some fairly close approximations can be made. The net result of this study will be a basis for several kinds of determinations that then need to be made.

First, in considering your production, think in terms of both peacetime and wartime conditions to determine whether your facilities might be or become a major military objective. This, in many instances, is comparatively easy to determine. A large aircraft plant, a shipyard, a large aluminum plant, each obviously could well become very important to an enemy. Generally, prime contractors for the production of war materials, in varying degrees of importance, could consider themselves in this class. Certain basic material plants and many component plants also would be included. It is not necessary to be a military expert to have a fairly good idea of the importance of your production in a war effort.

In making this determination, the size of your facilities and others within the area concerned should not be the sole influence in your decision. A small instrument plant, for instance, might be highly critical and the object of much attention from the strategic planner. Likewise, the fact that your industry is a heavy or a light industry has little bearing. The real determining factors are the actual use our military forces will make of the items produced, and the amounts of such items which are available in the proper places at a given time, as compared with the actual requirements, and the possibilities for employing substitutes. From this inquiry, you should be able to estimate whether or not, at some future critical moment, the enemy strategic planner might consider your facilities, or the area in which they are located, as a major objective. Even if you believe your facilities alone would not be classed as a primary objective, the existence of any other facilities within this three-mile radius, which could be so classified, is as much a matter of your concern as if your own facilities were directly involved.

A further inquiry into your facilities may show that all of them are not necessarily attractive targets. It may be that only a single unit, if destroyed, would result in a 100 percent stoppage of production for several months. This might be adequate to the needs of the enemy strategic planner. This means that all such units should be identified if you are to have a complete picture. While, of course, no attempt is likely to be made to knock out a single plant with an atomic bomb, conventional bombing techniques have improved to a point where such destruction is easily possible.

As a result of your investigation, you should be able to conclude for strategic vulnerability reasons, the following:

(*a*) Whether all of your facilities should be relocated in some other general area, or

(*b*) Whether part of your facilities should be relocated, or

(*c*) Whether no further expansion should be attempted in the present area.

You May Be Surprised.

An appraisal of the relative strategic position of your plant installations employing the guide rule described may produce seemingly startling results. If removal of part or all of your facilities is indicated, it is not recommended this be attempted at once. This probably would not be economically practical. Moreover, this would give rise to many allied problems which must be resolved. It is suggested, however, that any ex-

Drafting boards in underground factory, Mosbach, Germany. The factory produced aircraft engines, had 16,000 feet of floor space.

pansion of facilities in a location with a substantial degree of vulnerability be avoided .

It is also suggested that efforts be directed to the development of a progressive dispersion plan built around normal expansion and obsolescence factors. In short, any solution to this problem must be economically feasible. A program of this type must of necessity fit into long-range planning. The important thing is to devote study and efforts with a view to making the earliest possible start toward dispersion as the means of achieving a reasonable degree of plant security.

Selecting New Plant Locations.

Studies made to date indicate that areas of industry concentration less than five square miles, or urban concentrations of less than 50,000 people, separated by about 10 miles of relatively open country, will be reasonably secure from attack under all circumstances expected to prevail. These factors may be employed as a yardstick in evaluating prospective sites, although, admittedly, such thumb-rules tend to oversimplify the problem.

Among other things, your thinking should be projected in terms of the probable growth or foreseeable developments in the prospective area which may make it strategically vulnerable. It should also be borne in mind that the location of one industry in a community normally influences growth of other dependent or allied industries and services. General trends in population movement must also be given proper weight.

In this connection, topographical features of terrain become significant. In World War II, Hiroshima and Nagasaki were chosen as targets for the atomic bomb because of their strategic concentration of industrial activities. Deaths in Nagasaki, however, dispite its greater population density, were only one-half those at Hiroshima. Topography was the major factor in accounting for this difference. Ridges of hills separated areas of Nagasaki into dispersed built-up pockets, whereas Hiroshima was relatively flat and uniformly concentrated. The Nagasaki bomb thus dissipated much of its energy against hills and unoccupied area, while the Hiroshima bomb achieved its most devastating effect.

Industry fared better in Hiroshima. All major factories were reasonably well dispersed and escaped serious damage. At Nagasaki, plants and dockyards at the southern end of the city were left intact, but those con-

centrated in the valley where the bomb exploded were almost completely destroyed.

It is fully recognized, of course, that a fundamental requirement in any survey of an area for selection of a location is that it be economically feasible to the particular industry concerned. With few exceptions, however, economic advantage and relative strategic safety will not be incompatible.

Some of the basic factors governing the economic evaluation of plant locations are:

1. Location of production materials.
2. Labor.
3. Sites.
4. Industrial fuel.
5. Transportation facilities.
6. Market.
7. Distribution facilities.
8. Power and water.
9. Living conditions.
10. Laws and regulations.
11. Tax structure.
12. Climate.

All industries are dependent in some degree upon one or more of these factors in evaluating prospective locations. Industries least dependent upon these requirements for economic advantage will have the maximum latitude in site selection. Some, by the same token, will have little latitude.

When a comprehensive survey of prospective locations has been made, it is probable that several will be found to be economically practical. Your problem then becomes one of selecting the location which is strategically most desirable, employing the factors previously described.

Dispersed vs Underground Locations.

Adequate dispersion is the most practical solution to the problem of strategic location.

Any solution must reflect a judicious balance between economic feasibility and reasonable strategic security. Dispersion by and large meets these requirements.

Drill press bay in an underground V-1 factory in France. The factory employed 8000 workers. Its site was an old iron mine.

Underground installations, although probably providing maximum protection, are not thought practical for application on a large scale. Among other things, they generally entail higher construction costs. This is also true of special construction of above ground facilities, employing fire, blast, or radiation resistant materials. Resort to this type of security will probably be economically justifiable only in connection with such highly strategic facilities as may be determined by the Armed Forces.

Dispersion—Product and Plant.

In planning for the development of

a dispersion plan for your industry, an attempt should be made to disperse not only a maximum number of your manufacturing installations, but also the manufacture of individual products. In short, such a plan should be as broad as practicable, both plant-wise and product-wise, and designed to minimize interdependence; or stated another way, to achieve maximum self-sufficiency.

The degree of latitude in achieving product and plant dispersion, just as with geographic location, will of course vary with each industry.

Trends Already Visible.

Decentralization of population and industry is actually taking place. A host of pressures is at work in this direction.

The National Industrial Conference Board in its study, "Decentralization of Industry," states:

"There is a trend toward locating manufacturing plants in the smaller cities and towns. Cities and towns with 10,000 to 100,000 population are reported to be the most popular places for plants established from 1940-1947. Only one-third of the plants built or acquired since 1940 are in cities of 100,000 and over. For plants established prior to 1940, close to half were in cities of that size. On the other hand, almost 30 per cent of the plants established since 1940 are in towns of 10,000 or less, against only 20 per cent of the plants built before 1940."

Trends toward decentralization of industry and population are significant for several reasons. Each of these trends influences the further development of the other: Markets grow up in new areas. They in turn attract industry. New industry gives rise to new pools of highly skilled labor. This cycle of action and reaction sets up a benign circle.

Many industries are very dependent upon skilled labor, which is normally thought to be relatively immobile. Further developments in decentralization of population plus the growing development in simplified manufacturing techniques, semiautomatic machine tools, etc., go far toward reducing dependence upon localized supply of highly skilled labor. To these factors should be added the incentive to movement that can be created for workers in the form of better working and living conditions.

It may also be anticipated that agencies lending money for long-range capital expenditures will give thought to location as a risk factor. Thus, decentralization may be given further impetus.

These are a few of the many factors bearing a relation to the relocation problem. They indicate the broad outlines of a potential pattern of industry and population which, if properly guided, will result in substantial benefits to both.

It remains industry's problem to take maximum advantage of this pattern to insure that new plants are located with a view to maximum security. The fact that your particular facilities are decentralized is not sufficient if the bulk of your installations, although dispersed, are located in highly vulnerable areas.

Conclusions.

The whole problem of industrial dispersion boils down to a common sense application of the old adage about not putting all of one's eggs in one basket.

Whatever future technological advances may take place in the art of waging war, there will always remain the fundamental principle that the advantage sought in any strategic objective must be weighed against the cost of attaining it, in terms of energy and materials. It is also true that as the destructive effects of modern weapons increase, so does their cost. Such weapons—whether they be atomic, chemical, bacteriological, or any other diabolical engine of destruction that may be further developed in the fore-

seeable future—cannot be expended prodigally unless the risk is calculated to bring commensurate results.

If the industrial facilities of the United States were effectively dispersed, that fact alone would make an incalculable contribution toward the maintenance of peace because of the prohibitive expense of any enemy attempt to destroy this country's ability to defend itself. Dispersion could contribute significantly toward outlawing war.

The job of dispersion is one that industry must assume, for both its own protection and that of the national security. Ours being a democratic Nation dedicated to the principles of free enterprise, the Government can neither dictate nor finance such a large-scale change in the industrial pattern. While the Government is naturally concerned with the promotion of a healthy condition in private enterprise, it also has a primary concern for the security of the Nation as a whole.

As technological developments advance to meet changing conditions in the rapid unfolding of world events, many of the problems which now appear nebulous are certain to come into sharper focus. Effective solutions to these problems will be vastly facilitated by the combined and cooperative efforts of industry and Government.

QUALITATIVE ANALYSIS OF FOLDED DIPOLES.

ACKNOWLEDGMENT.

This timely article of interest in connection with the current television vogue was written by LIEUTENANT COMMANDER CHARLES W. HARRISON, U.S.N. It was originally published in the March, 1949 issue of "Buships Electron."

Introduction.

Folded dipoles are receiving wide usage at the present time for two basic reasons: First, the input resistance of a two- or three-wire folded dipole at resonance falls within the limits of 250 to 300 ohms or 600 to 650 ohms, respectively. It is entirely feasible to construct an ordinary air dielectric two-wire transmission line having a characteristic resistance of any value within the range 250 to 650 ohms. This fact obviates the necessity for employment of transmission line sections as impedance transformers for matching the antenna to the feed line. Second, a folded dipole used at the lower communication frequencies exhibits a more satisfactory impedance against frequency characteristic than a simple symmetrical center-driven antenna of comparable overall length and wire radius.

The present article presents a qualitative analysis of folded dipole operation. A formula for roughly estimating the input impedance of a folded dipole consisting of N wires is developed.

Two-Element Folded Dipole.

A conventional two-wire folded dipole is shown in Figure 1. The wires are of the same radius, and their length approximates one half wavelength. For convenience in this and in subsequent figures the elements are shown unduly separated; however, it is to be understood that the distance between wires is small compared to the wavelength

Figure 1.

Conventional form of folded dipole, showing driving transmission line.

Figure 2 illustrates a folded unipole. The lower half of the two-element folded dipole is replaced by a large perfectly conducting plane surface. A source of voltage is shown connected between the lower end of one wire and ground. Provided no auxiliary conductors or dielectrics are located in the vicinity of either the folded dipole or folded unipole, the input impedance of the folded dipole is precisely twice the input impedance of the folded unipole.

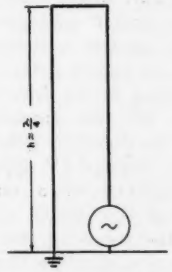


Figure 2.

Folded unipole. The lower half of the antenna shown in figure 1 is replaced by a large perfectly conducting plane.

Figure 3.

Base-driven vertical radiator over a large perfectly conducting plane.

A quarter-wave base-driven vertical radiator is shown in Figure 3. The radius of wire employed is the same as the radius of wire used in the construction of the folded unipole. It is desired to establish a relationship between the driving point impedance of the folded unipole and the vertical radiator. For this purpose it is convenient to replace the single generator shown in Figure 2 by the three identical impedanceless generators shown in Figure 4. The polarities of the generators at a selected instant in time are indicated. The fact that identical impedanceless generators are not physically realizable is of no import; they are introduced for purposes of discussion only and play no significant part in the final result. The "generator region" of Figure 4 and in subsequent figures is deliberately exaggerated to insure space for appropriate labeling of the drawings.

Assume for the moment that no voltage is developed by generator 3. In this case a voltage 2V appears across the sending end of a quarter-wave transmission line which is short circuited at the receiving end. The input impedance of such a line section is very high if it is adjusted to resonance, and accordingly very little current flows into the line at the driving point. However, large currents may exist at other points, particularly in the region of the shorting bar between wires located at the receiving end of the line. Actually, these currents are sinusoidally distributed along the line to a high degree of approximation. They play little part in radiation from the antenna because, at any given distance along the wires, they are of equal amplitude but flow in opposite directions. This is defined as the antisymmetrical mode for currents.

Next, assume that generators 1 and 2 develop no voltage. The unipole now resembles the base driven vertical radiator of Figure 3 except that, in lieu of one wire, two wires form the vertical section. Because of symmetry, the total current divides equally between the two elements. Notice that at any given distance along the wires the currents are of equal amplitude and flow in the same direction. This is defined as the symmetrical mode for currents. They play a dominant role in radiation from the antenna.

Suppose, now, that all generators are operating. It is immediately apparent that the left element is grounded at its lower extremity because it is connected to ground through two identical impedanceless generators whose voltages oppose. On the other hand, the voltages of generators 2 and 3 develop a voltage

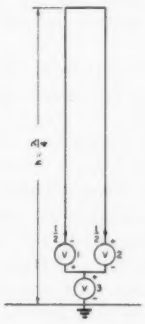


Figure 4.

Folded unipole like figure 2. The single generator of figure 2 is replaced by three identical impedanceless generators, as shown.

2V between the lower extremity of the right-hand element and ground. The input impedance of a folded unipole is, therefore, $2V/(I/2) = 4V/I$. This result may be correlated with the input impedance of the quarter-wave vertical radiator, it being presumed that wire of the same radius is employed in constructing both, and that the overall length of the vertical radiator is the same as the distance between the driving point and center of the shorting bar of the folded unipole. Obviously, the driving point impedance of the vertical radiator is V/I. Accordingly, the impedance of a folded unipole is approximately four times the impedance of the comparable base-driven vertical radiator. (Observe that for the symmetrical mode for currents the presence of the shorting bar does not significantly alter the current distribution along the wires inasmuch as it is connected between points of equal potential. This is not true for the anti-symmetrical mode for currents.)

The limiting value for the input impedance of a base-driven quarter-wave vertical radiator (as the radius of wire approaches zero) is $36.57 + j21.25$ ohms. Therefore, the input impedance of a folded unipole comprising two elements is $146.26 + j85$ ohms. The input impedance of an isolated center-driven half-wave antenna of vanishingly small radius is $73.13 + j42.5$ ohms. Accordingly, the input impedance of an isolated folded dipole composed of extremely thin wires is $292.5 + j170$ ohms.

Ordinarily the input impedance of a folded dipole (or folded unipole) with air dielectric is complex; i.e., consists of a reactive as well as resistance component. The total current at the driving point is a superposition of the symmetrical and antisymmetrical currents. It might prove difficult to obtain antenna resonance as well as transmission line resonance simultaneously. Antenna resonance is achieved when the symmetrical component of current entering the driving point (which, except at resonance, is complex *per se*) is precisely in time phase with the applied voltage. Transmission line resonance requires the antisymmetrical component of current to be vanishingly small at the driving point. Undoubtedly, a judicious choice of wire spacing and wire radius is required if one contemplates obtaining a purely resistive input impedance.

It has become common practice to construct folded dipoles resembling Figure 1 from twin-lead transmission line, the wires being secured along the edges of polyethylene ribbon. Possibly, the wavelength associated with the antisymmetrical mode is somewhat shorter than that associated with the symmetrical mode under these circumstances.[1] Based on this premise a quarter-wave transmission line section would be somewhat shorter than the section of wire required for antenna resonance. Accordingly, if the input impedance is to be a pure resistance, two adjustments are necessary. First, a rigorous version of the theory of coupled antennas, suitably modified to include the effect of a thin dielectric film surrounding the wires, should be employed for estimating the length of wire required to obtain antenna resonance. Second, the location of the shorting bar for obtaining transmission line resonance must be computed by use of a modified transmission line theory which takes cognizance of the thin dielectric strip between wires.[2]

[1] The available theory of biconical antennas having very small angle cones with space between them filled with suitable dielectric material should prove efficacious for estimating the effect of a thin dielectric film on the resonant length of an antenna. An experimental investigation will be made later to determine the difference between the wavelengths associated with the symmetrical and antisymmetrical modes for wires embeded in thin dielectric ribbon. This will be the subject of an article to be published later.

[2] A transmission line theory for wires immersed in solid dielectric media (extending throughout all space) is available. One should exercise caution in applying the results of this theory to the case of wires embeded in a thin dielectric ribbon. The two cases bear little geometrical similitude.

If the wave-length associated with the symmetrical component of current actually exceeds the wavelength associated with the antisymmetrical component of current as expected, then one way out of the dilemma is to provide for an adjustable shorting bar. An illustration of the resulting "folded dipole", which has now become a bridged-T antenna, is shown in Figure 5 It is evident that the length of line section S necessary to obtain transmission line resonance (measured from the driving point to the center of the shorting bar), and the length of antenna 2h necessary to obtain antenna resonance, may be determined experimentally.

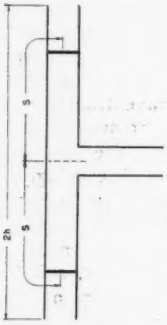


Figure 5.

Bridged-T antenna. The length 2h is adjusted to obtain antenna resonance. The length S is adjusted to secure transmission line resonance. The wires may be embeded in dielectric media other than air.

A folded dipole antenna inherently possesses a more satisfactory impedance against frequency characteristic than a simple dipole of comparable wire radius. For purposes of explanation, assume that at a given frequency the folded dipole is precisely resonant. If the frequency of the applied voltage is lowered (the wave-length increased) the antenna is too short and the input impedance is capacitive in character; i. e., the symmetrical component of current leads the applied voltage. Simultaneously, the shorted line section becomes too short and the input impedance is inductive in character; i. e., the antisymmetrical component of current lags the applied voltage. The resulting driving point current, which is the vector sum of the two components, remains fairly well in phase with the applied voltage. If the frequency is raised, the symmetrical component of current lags, and the antisymmetrical component of current leads the applied voltage. The reactive parts of the two components of current again tend to compensate one another and the total input current remains fairly well in phase with the applied voltage. Thus the impedance against frequency characteristic of a folded dipole is reasonably smooth over a band of frequencies on either side of resonance.

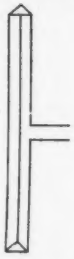


Figure 6.

Three-wire folded dipole. The wires are secured at the vertices of metallic equilateral triangles.

Three-Element Folded Dipole.

Figure 6 illustrates a three-element folded dipole. The wires are approximately one half wavelength long, and their ends are secured at the vertices of metallic equilateral triangles, as shown. The assumption is made that all wires are of the same radius. One wire of the three-wire antenna is

Figure 7.

Three-wire folded unipole. The lower half of the antenna shown in figure 6 is replaced by a large perfectly conducting plane.

center-driven; the other two are parasitic. Figure 7 portrays a three-element folded unipole. The lower half of the three-wire folded dipole is replaced by the earth. The wires are in the vicinity of one quarter wavelength in length. The antenna is driven by connecting a generator between the lower extremity of one wire and the ground plane.

In order to provide a qualitative explanation of the operation of the three-element folded unipole, the single generator of Figure 7 is replaced by seven identical impedanceless generators shown in Figure 8. A modified version of the theory of symmetrical phase components is employed. By use of the superposition theorem the current entering the driving point of Figure 7, or the right-hand wire of Figure 8, is determined, and from it the driving point impedance.

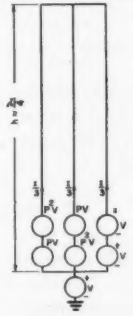


Figure 8.

Three element folded unipole, showing driving generators. The method of analysis is based on the theory of symmetrical phase components.

The polarities of the various generators at a selected instant in time are indicated. In this case the phase factor P means that as one progresses along a horizontal row of generators (always in the same direction) a progressive phase angle of 120 degrees is introduced between the applied voltages. If the top layer of generators drives the wires in the positive sequence, the next layer of generators drives the wires in the negative sequence.

As mentioned earlier, all applied voltages are of the same amplitude, but considering one layer of generators alone an appropriate phase angle is introduced between the voltages to insure maintenance of electrical symmetry in the three-element antenna. Symmetry cannot be accomplished unless the phase factor P satisfies the equation

$$P^2 + P + 1 = 0 \quad \ldots\ldots\ldots\ldots\ldots\ldots (1a)$$

or

$$P^2 + P = -1 \quad \ldots\ldots\ldots\ldots\ldots\ldots (1b)$$

Multiplying through by V gives

$$P^2V + PV = -V \quad \ldots\ldots\ldots\ldots\ldots (1c)$$

Thus the two generators in the left-hand wire of Figure 8 may be replaced by a single generator of voltage V having polarity opposite to the polarity of the generators located in the right-hand wire. Similarly, the two generators in the center wire are replaced by a single generator. Figure 9, involving five generators, is thus electrically equivalent to Figure 8 where seven generators are required. One might say that the three-phase problem pictured by Figure 8 has been replaced by the single-phase problem pictured in Figure 9. The argument is now the same as that presented for the case of the two-wire folded unipole.

If the generator connected between the earth and the antenna is operative and the others are not, a current I/3

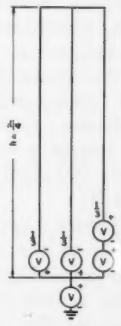


Figure 9.

Three-element folded unipole being driven by a generator system electrically equivalent to the system pictured in figure 8.

will exist in each wire. This is true provided geometrical symmetry obtains; i. e., the wires are equally spaced on the circumference of a circle. For any other orientation of wires it is not correct to assume that the current divides equally between the wires, as the coupling between them is no longer equal. When the upper generators act alone essentially no current flows in any wire provided the length of the wires is adjusted to achieve transmission line resonance. When all generators are operating, the center wire and left-hand wire are effectively base grounded since the impedanceless generators are connected such that their equal instantaneous voltages oppose. Inspection of the figure reveals that a voltage 3V acts between ground and the lower extremity of the right-hand wire. The input impedance of a three-element folded unipole is therefore $3V/(I/3) = 9V/I$. If one assumes that the input impedance of a quarter-wave vertical radiator is $36.57 + j21.25$ ohms, the input impedance of a three-wire folded unipole is approximately $329.1 + j191.3$ ohms. Considering that the input impedance of a three-element folded dipole is precisely twice the input impedance of the equivalent three-element folded unipole, one obtains $658.2 + j382.5$ ohms as the input impedance of an isolated three-wire folded dipole. If adjustments are made to insure simultaneous resonance with respect to the symmetrical and anti-symmetrical components of current, the three-element folded dipole described will present a resistive load of approximately 600 ohms to the driving transmission line.

N-Element Folded Dipole.

An analysis of the operation of a folded dipole consisting of N identical wires equally spaced about the periphery of two metallic rings, one wire being center-driven, proceeds as follows: Replace the N-element folded dipole by an N-element folded unipole located over a large perfectly conducting surface. The antenna is driven by a single generator connected between the lower extremity of one wire and ground. Next, a system of idealized generators is introduced in lieu of the single driving generator. The number of them and their phase relationship is readily

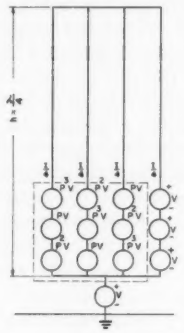


Figure 10.
Four-wire folded unipole showing driving generators.

established, as indicated below. For each horizontal row of generators the phase factor P must always satisfy the equation

$$\sum_{q=0}^{n} P^q = 0 \qquad (2)$$

for electrical symmetry to obtain. In (2) $n = N - 1$. Also the number of phase sequences required (and hence the number of generators needed in each vertical wire) is n. Observe that

$$\sum_{q=1}^{n} P^q = -1 \qquad (3)$$

so that the vector sum of the voltages effective in each wire (with the exception of the wire containing n generators operating in the same phase) is always the voltage of a single generator having polarity opposite to the generator establishing the reference for phase in the system. Accordingly, the N phase problem is reduced to a single phase problem. Reference is made to Figure 10. Here $N = 4$ and

$n = 3$. Obviously a total of $N^2 - N + 1$ $(= 13)$ generators are needed. The equivalent arrangement of Figure 11 requires $2N - 1$ $(= 7)$ generators. For this case the input impedance (by inspection) is

$$NV/(I/N) = N^2 \frac{V}{I} \left(= 16\, V/I \right)$$

The formula for estimating the input impedance of an N-element folded unipole (or N-element folded dipole) is therefore

$$Z_{in} = N^2\, V/I \qquad (4)$$

The ratio V/I is usually taken as 36.57 + j21.25 ohms for an N-element folded unipole, and 73.13 + j42.5 ohms for an N-element folded dipole.

Readers are reminded that (4) is an approximate formula only. Relatively large errors in calculating the input impedance of N-element folded antennas may result when $N > 3$. Also the ratio V/I depends in a very fundamental way on the number of wires involved, on their length and radius, and the gap width at the driving point.

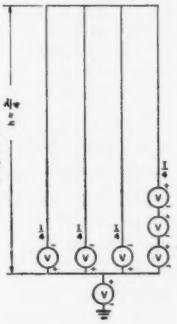


Figure 11.

Four-wire folded unipole being driven by a generator system electrically equivalent to the system pictured in figure 10.

PILOT HOUSE CONTROLS.

ACKNOWLEDGMENT.

This article is reprinted from "Motorship" for April, 1949. It describes several types of pilot house controls which have been developed in answer to the demand on the part of operators for improved maneuverability of vessels operating in inland waterways.

In the days when sail was the main propulsion power on the sea, the man at the wheel had the whole main deck under his gaze, could issue his commands to the men in the rigging, and see his commands executed. As sail yielded to power propulsion, the main propulsion force on the ship began to move below decks, and the helmsman's position was moved forward until he was quite out of sight and sound of the crew who were in direct control of the ship's propulsion.

The system evolved by which the pilot communicated with his engine room was the gong and jingle bell, later replaced by the mechanical telegraph. Several codes of signals were employed, but the one commonly used on salt water was one gong—slow ahead or stop, two gongs—slow astern. The jingle bell was sounded for full speed ahead or astern after the gong signal. It was also used for stand-by and ring-off signals. Variations were numerous, such as jingle-gong for dead slow or double jingle and gong for slow as possible.

The mechanical telegraph, a later development, offered a combination of audible and visual signals. This, as everyone knows, is a lever mounted on a pedestal and usually chain-connected to a similar lever in the engine room. Operated manually, it communicates orders from the bridge to the engine room and permits acknowledgments from the engine room of orders received.

Thus, when the pilot required a change in the operation of his engines, he signalled his order on the telegraph. This was received in the engine room, executed, and acknowledged by the engineer or his assistant. The system, while quite efficient in unhurried operations, has definite limitations, most evident in rapid maneuvering and allows of usually no more than three gradations of speed forward or astern.

In slow moving ships, the pilot is allowed a greater span of time "between the wish and the fulfillment," and the fact that he does not have direct control of the ship's engines is not too important. However, as the speed of the vessel increases, the time allowable for the execution of a maneuver becomes less and less until a point is reached where, in heavy harbor traffic for example, anything short of instantaneous execution of engine maneuvers can be dangerous.

No ship's telegraph system, of course, can provide instantaneous execution of pilothouse commands. As a result, there have been developed systems of pilothouse controls, which through various forms of linkages, afford the pilot direct control of the engines, enabling him, in effect, to execute his own orders instantly.

Obvious advantages are gained by putting full control of a vessel under the hands of her pilot, and pilothouse control, originally a feature found only in small boats, has been extended to in-

clude vessels in the big ship class. It is accepted as standard on newbuilding river and harbor craft, and the resultant advantages have assured its retention in marine construction.

Generally speaking, in geared diesel-powered vessels, pilothouse controls must afford the helmsman direct control over the engine's throttle, permitting him to vary engine speed, and over the clutch, permitting him to shift gears to change the rotation of the ship's propeller.

As, in a vessel of any size, the pilot is considerably removed from both the engine's throttle and clutch lever, a system of linkage must be installed between the pilot's position and the engine room that will transmit energy from the pilot's hand to the throttle and clutch.

Linkages of almost any type or combination of types can be used to transmit control actions, and systems are available on the market to fulfill any requirement for pilothouse control.

Most common in the types of linkages used in pilothouse control of geared diesel-powered vessels are perhaps (1) mechanical, (2) hydraulic, (3) electric, and combinations of these three.

Mechanical Linkage.

Mechanical linkage, as the term implies, indicates a direct connection by chain, rods or flexible cable between pilothouse controls and the engine and gear. Simplest of all the types, it offers a high measure of dependability and positive control. There are, however, some limitations connected with the system. It cannot be used, for example, over too great distances, where the inertia of the linkage itself would be difficult to overcome in adjusting the controls. Obstructions in the linkage path would be another barrier to its use.

However, in situations where the pilot station is not too far removed from the engine room and where a reasonably direct, unobstructed path for the linkage is available, the mechanical type of linkage is highly satisfactory and trouble-free.

Typical of the mechanical type linkage controls available on the marine market is the Magicam Control developed by the Columbian Bronze Corporation of Freeport, Long Island. This is a combined mechanical clutch and throttle control for engines with hydraulic clutches. This device, which finds its broadest application in the commercial workboat fields, offers the pilot control of both throttle and reverse gear in a single control handle. A safety feature incorporated in its design prevents operation of reverse gear at other than idle engine speed, thus protecting the ship's propulsion equipment from damage through careless or inexperienced handling. A clutch disengage knob on the handle allows adjustment of the throttle alone. The control handle, through a cam action, moves clutch and throttle levers mounted on the side of the control casing. These are direct-connected by rods and bellcranks as required

The use of flexible cable in mechanical linkages permits by-passing of some obstructions that would preclude the use of rods, but a certain amount of discretion must be used in installation to avoid unnecessary bends and to keep the required bends as large as possible.

The well-known Panish Controls organization of Bridgeport, Connecticut, have on the market a widely accepted system that offers mechanical linkage by flexible cable. The Panish product is designated as the Positrol and is a combination clutch and throttle control, particularly suitable for engines with hydraulic reverse gears. Although there are two separate control levers, they are so mounted that both may be gripped easily with one hand and operated simultaneously to ahead, neutral, and reverse positions.

An optional feature with the Positrol is a mechanical interlocking system which will prevent opening the throttle before the clutch is engaged, or disengaging the clutch when throttles are opened beyond idling position.

It is claimed that the Positrol is unusually simple to install and that the flexible push-pull cables transmitting the control action are no more difficult to position than ordinary wire. Combustion governor and remote engine stop controls are a part of the Positrol system when used with diesel engines.

The flexible cable system of control has been referred to quite commonly as the "push-pull" type and is well-suited for the job of both throttle and clutch control where the engine's gears are engaged by hydraulic force, requiring only a minimum pressure on the clutch control lever on the gear housing.

The products of Simmonds Aerocessories, Inc., of New York City, Arens Controls, Inc., of Chicago, and the Le Bon-Morris Company of Los Angeles belong to the same general family of mechanical linkage controls, and each employs flexible cable on the "push-pull" principle.

Another of the same species is the single-lever combined engine and throttle control developed by the Morse Instrument Company of Hudson, Ohio, and similarly linked to the engine and reverse gear by flexible cable.

Mechanical linkage permits of many variations, and the control in the pilothouse may take the form of a wheel, lever, foot treadle, or any other convenient arrangement permitting the transmission of linear motion.

In certain situations, where the control stand is not too remote from the engine room and only a limited torque is required to move the clutch lever, controls of the mechanical linkage type have the definite advantage of simplicity, positive action, and minimum maintenance requirements.

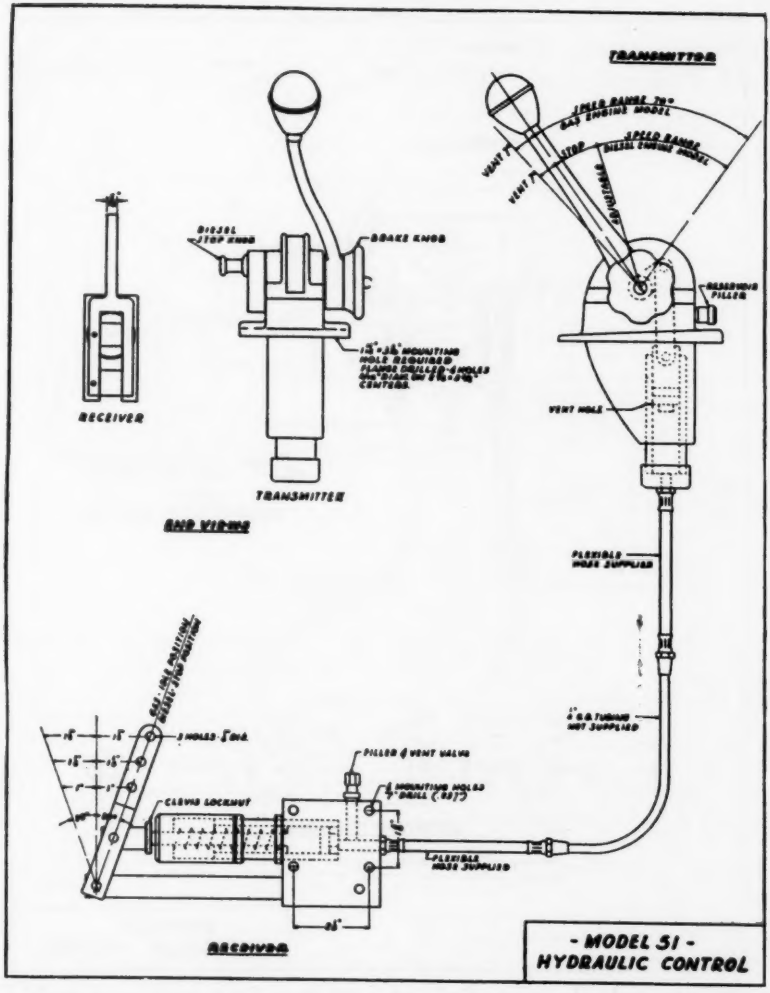
Hydraulic Linkage.

In situations where the remote control station is located at a considerable distance from the engine room and/or the torque required to move the gear lever is much higher, the installation of a hydraulic system may be in order.

The effectiveness of hydraulic linkages stems from Pascal's Law of the transmissibility of pressure, whereby changes in pressure are transmitted equally throughout a liquid. Thus pressure applied at one end of a tube filled with liquid will be transmitted equally throughout the liquid and be manifested at the opposite end of the tube. In practical application movement of a controlled piston at one end of a liquid-filled tube will cause a corresponding movement in a free piston at the opposite end, assuming a condition of perfect seal. Thus a simple remote throttle or clutch control embodying hydraulic linkage has the remote control lever connected to a piston operating in an oil-filled cylinder connected by tubing to a "receiving" cylinder. Linear motion is transmitted through the liquid in the tube and activates the free or "receiving" piston attached to the engine's throttle or reverse gear lever.

The Model 51 hydraulic throttle control developed by the Columbian Bronze Corporation is a good example of a simple hydraulic linkage. The transmitting unit is a single lever mounted at the pilothouse control station. The lever is direct-connected to a small piston. Any linear movement of the transmitting piston is reproduced by the springloaded receiving piston, which in turn is direct-connected to the engine's throttle lever.

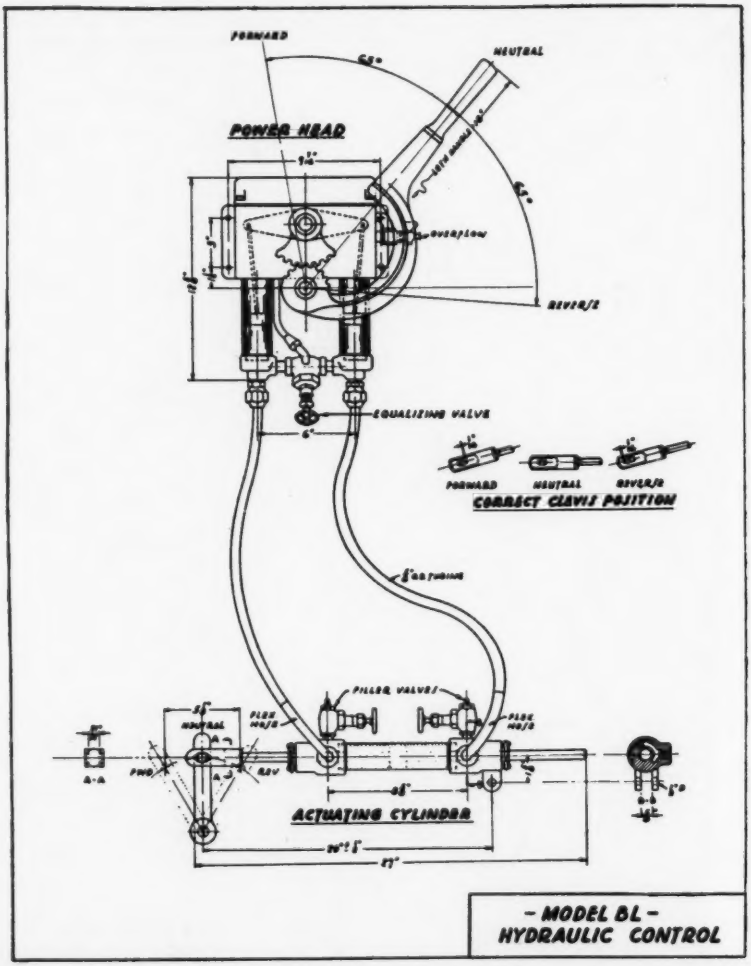
A hydraulic linkage does not impose any limit on the distance covered nor on the path taken by the linking tubes, as it is the liquid that is the medium of transmitting motion. However, care must be exercised in keeping the lines tightly sealed, as leakage would obviously render the system ineffective.



Columbian's single line spring-loaded hydraulic system is designed principally for engine throttle control. A double line system, opposite, is used for gear control.

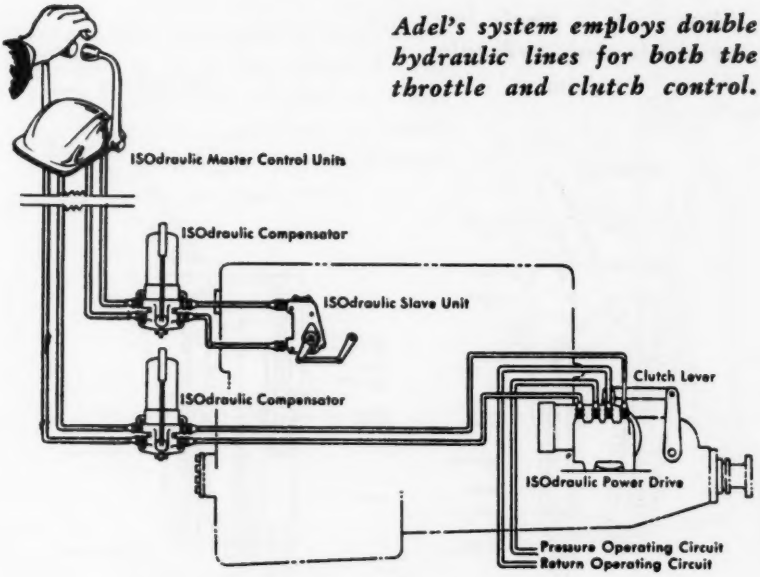
In addition, changes in fluid volume caused by temperature differential must be compensated for in order to keep both pistons in relative alignment in their respective cylinders. However, most systems are designed to compensate for this latter factor automatically.

Manually controlled reverse gears generally require considerable torque to move them, and a hydraulic system to



Illustrated above is Columbian's two-line system capable of transmitting the greater torques required for gears.

afford clutch control should be of more rugged design than that for throttle control. Columbian's Model BL hydraulic clutch control is a two-line system that can transmit pressures against a clutch lever equivalent to torques up to 2000 in. lb. in either the forward or astern direction. The control handle is geared to a rocker arm. Connecting rods at either end of the rocker arm move two pistons in separate cylinders. The gearing provides a ratio of 1 to 6¼ of control handle force to piston rod force.



Adel's system employs double hydraulic lines for both the throttle and clutch control.

The receiving unit is a double-acting cylinder carrying a piston whose rod is direct-connected to the gear shift lever. By this arrangement and that of the transmitting power head, the operator may bring considerable force to bear on the clutch lever in either a forward or reverse direction.

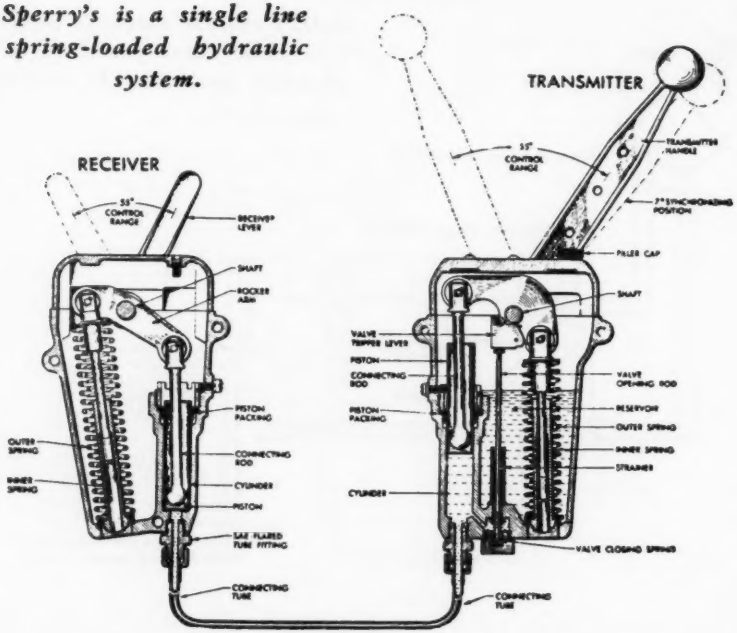
Another well-known product in the field of hydraulic linkage controls is marketed as the ISOdraulic control system by Adel Precision Products Corporation of Burbank, California.

The essential units comprising this system are: a manually operated master control unit; a temperature equalizer or pressure compensator; a remote unit for translating pressure energy into motion; and of course a circuit of tubing by which pressurized hydraulic fluid connects the units in the system.

The ISOdraulic system is designed for engine throttle control and similar applications where loads are usually in the neighborhood of 25 in. lb. and not in excess of 100 in. lb. In proper installations an unusually high degree of precision and operating sensitivity has been demonstrated by this system. When used in conjunction with a power-driven system for operating the clutch, it is claimed that the high sensitive transmission characteristic enables the operator at remote position to feel the position of the clutch just as if he were actuating the clutch lever in the engine room.

The manufacturer regards the ISOdraulic system as being of two basic types. The first transmits working force from the master unit to slave unit in one to one ratio less system friction. The second basic type transmits control energy from the master unit to the slave unit, which in turn controls the full power of a pressure-operated hydraulic cylinder assuming loads up to 10,000 in. lb. Both types, it is claimed, offer a positive control of movement and position. The ISOdraulic

Sperry's is a single line spring-loaded hydraulic system.



is a two-line system, transmitting a load of equal magnitude in either direction without lag.

A single-line hydraulic control system has been developed by Sperry Products, Inc., of Danbury, Connecticut. This is marketed as the Sperry Exactor hydraulic control and has a capacity of up to 400 in. lb. The system consists essentially of a transmitter and a receiver, both incorporating similar trunk-type pistons and spring arrangements, the latter serving to preload the system in such a manner that any movement of the transmitter handle is communicated immediately and positively to the receiver lever arm.

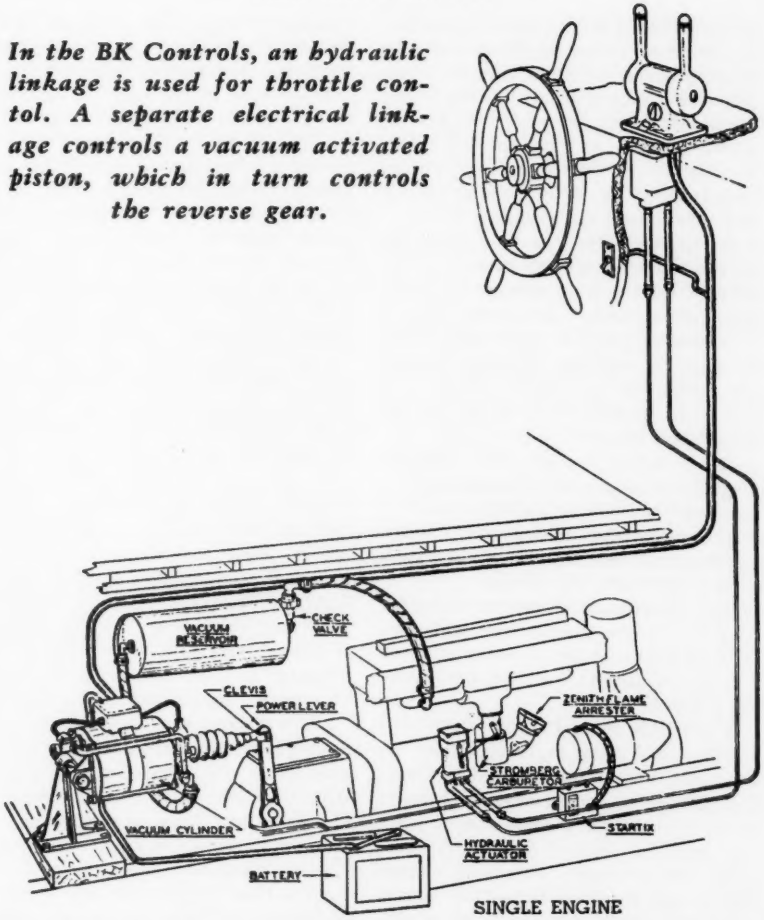
Built into the transmitter unit of the Sperry system is an oil reservoir and valve that permits compensation to be made for changes in fluid volume through temperature changes. The receiver spring can be so adjusted that in cases of accidental fracture releasing pressure in the system, the receiver lever will automatically move to a predetermined safety position, either full open or full closed.

One of the more obvious advantages claimed for the one-line system of hydraulic linkage is of course simplicity of installation.

Another hydraulic linkage of the two-line hydraulic system is manufactured by the Marine Equipment Division of Ellinwood Industries of Los Angeles, California, and consists like the others of master control unit, compensator, and slave unit. It is claimed by the manufacturer that up to 500 in. lb. of torque may be transmitted by Ellinwood hydraulic controls.

In general practice, linear motion, such as control action, is transmitted from remote positions through mechan-

In the BK Controls, an hydraulic linkage is used for throttle contol. A separate electrical linkage controls a vacuum activated piston, which in turn controls the reverse gear.



SINGLE ENGINE

ical or hydraulic linkage as described and exemplified above. By these means, motion at a transmitting unit is duplicated at a receiving unit.

However, effective remote control does not always require the transmission of exactly duplicated linear motion, and certain situations may require entirely different methods.

Electrical Linkage.

Control *energy,* rather than real linear motion, can be conveyed in the form of electrical impulses through a conductor and converted into control motion at the receiving end of the control circuit.

The BK Marine Remote Controls, formerly a Bendix product and now manufactured by the Conlan Electric Corporation of Brooklyn, New York, uses an electrical linkage in its gear control circuit. Conventional pedestal type controls in the pilothouse or bridge transmit an electrical impulse through a simple circuit to electrically-actuated valves mounted on a vacuum cylinder.

Moving within the cylinder is a double-acting piston linked to the reverse gear shifting lever. Constant vacuum is obtained in a tank evacuated by the intake manifold of a gasoline engine or vacuum pump in the case of diesel installation. A line from the vacuum tank is led to the valve housing, thence to either side of the cylinder.

It can be seen that it is only the valves that are directly linked (electrically) to the control station. As the forward or reverse valve is opened, the corresponding side of the double-acting cylinder is vented to vacuum, and the piston moves in instant response to this yielding force and throws the reverse gear lever.

As electrical linkage permits great simplification of controls, the pilothouse gear control can be reduced to three push buttons, one for forward, one for neutral, and one for astern; and such an arrangement is available.

The accompaying throttle control may be of the double hydraulic line type or a pneumatic, vacuum actuated linkage, venting to the same tank serving the gear shift.

The highly refined system developed by Panish, known as their Model 99, combines both electrical and mechanical linkage in remote clutch and throttle control. In the Panish system only one lever is required to furnish complete remote control over both engine clutch and throttle. By virtue of an excellent record of performance under wartime conditions, the Panish controls are well-recommended for constant heavy-duty operation in the larger commercial craft.

Basically, the system includes the remote control lever, electrical and mechanical linkages, reversing switch, reverse gear operating unit, and terminal throttle connection.

Engineering of a high order is involved in the design and operation of the Panish Model 99, and many unique features and safeguards have won the equipment high praise from many sources. Through a system designated "Torque Responsive Switching," marine reverse gears may be operated at high speeds without jamming, and it is claimed that full forward to full astern can be achieved in less than one second of elapsed time, and at the instant a maneuver is completed, all thrust load is removed. The torque output of the reverse gear operating unit is limited to a predetermined maximum, thus protecting both the gear and control mechanism against overload.

Signal buzzers built into the system sound while the reverse gear unit is in operation, going on as soon as the reverse gear has started to move, and ceasing to buzz as soon as the reverse gear has fully engaged in the reverse position. The buzzer is actually a signal that the throttle may be opened. In a normal maneuver the buzz will be of less than a second's duration, but a sustained, interrupted signal indicates a condition obstructing the engagement of the reverse gear. A continuous note indicates low battery. Both situations, of course, require immediate remedy.

The required force to shift the gear is provided by an electric motor which may draw the small current needed from the main engine starting battery. Heavy power lines are restricted to the engine room, the remote pilothouse circuit conducting only control impulses. This design embodies a wealth of protective features and advantages of operation that do much to establish pilothouse control as safe, dependable, and efficient.

Another system of control utilizing an electrical linkage is the Willis Remote Reverse Control manufactured by the E. J. Willis Co., of New York City. Control over the reverse gear is effected by a simple electrical circuit actuating valves in a double-acting cylinder which may be vented to vacuum or air pressure according to the model installed. Thus an electrical linkage controls

pneumatic force, either vacuum or compressed air, which in turn, through a double-acting piston, shifts the gears. The Willis Remote Reverse Control is used in conjunction with a throttle control of the mechanical linkage type.

Illustrative of the unusual forms remote controls can take is the single lever control system developed by the Morse Instrument Co., designed to centralize the control of steering as well as throttle and gear in one lever. In the Morse system hydraulic* power, generated by an engine-driven pump, is utilized to perform the work of steering, gear operation, and throttle setting. The main control lever is linked mechanically to hydraulic cylinders that deliver the force necessary to execute maneuvers.

The foregoing are given merely as illustrative examples of the usual types of linkages, some extremely simple, some highly engineered. The subject has been limited by confinement to the remote control of geared diesels. Diesel-electric propulsion and controllable-pitch propeller installations lend themselves as well, if not better, to direct control from the pilothouse. Some shipyards have developed their own systems, and certain large engine manufacturers supply complete pilothouse control equipment as part of their marine installations.

Pilothouse control has been established in actual operations as the most desirable arrangement in almost all types of commercial and pleasure boating. Judging from the broad variety of control systems that are available on today's marine market, there is no normal situation requiring remote control that cannot be served by any of the linkages described or combination thereof.

The controls described above by no means cover all the notable contributions of our manufacturers to the marine diesel field, and their mention only serves to indicate the broad diversity of types that have been developed in recent years

PROPULSION EQUIPMENT FOR 12,500 HP SUPER TANKERS.

ACKNOWLEDGMENT.

The two complementary articles which follow were prepared by MR. F. V. SMITH of the General Electric Co., and are reprinted from the "Pacific Marine Review" for January and February, 1949.

(PART ONE—JANUARY ISSUE).

The design engineers had a lot in the back of their minds when laying out the machinery arrangement for the new 12,500-Shp. tankers In the first place, they wanted to go completely modern and arrive at the absolute minimum fuel value possible without crossing over the borderline into what may be termed the area of "gold-plated" engineering. The trade route was long. On one side of the ledger appeared the favorable items—high speed and increased turn-arounds; and on the other side of the ledger, the unfavorable item of high fuel weight and cost.

Slow speed ships become valueless because of their limited turn-arounds, and high speed ships equally valueless because their deadweight cargo carrying capacity is taken up by the weight of the fuel. For a ship of any given size and tonnage, there is an optimum value between the two extremes which gives the best overall economic results.

Juggling the three factors—speed, fuel consumption, and ship's tonnage—to obtain a value which can be interpreted into terms of "minimum cost per ton mile for transporting cargo," requires a long series of calculations.

Great progress has been made in tanker design during the past twenty-five years. Ships have been made larger and speedier as more economical forms of propulsion apparatus have been devised. In the new ships the weight of the propulsion apparatus is being reduced to the irreducible minimum by selecting turbine-gear drive, which has the lightest weight per horsepower of any form of propulsion equipment. It is also the most economical form of turbine propulsion equipment because its transmission losses are low.

From a broad standpoint, overall economy in a steam power plant can be obtained by increasing the initial steam pressure and temperature on a progressive scale as the power rating of the equipment is increased; by including the regenerative method of heating the feed water with extraction steam; and by including a highly efficient auxiliary power plant.

The new ship are already being dubbed the "super tankers." When a tanker has a dead weight tonnage of 26,000 tons, carries 288,000 bbls, of cargo oil, makes 16 knots (or better), and carries a steam pressure of 850 lb.g. at a temperature of 850 F. in her boilers, it really proves that the designers have gone to town. When one studies the steam and feed cycle of the new ships, and the unique method of co-ordinating the various elements of design, it also readily becomes apparent that the bottom of the barrel has been scraped in this Btu. saving business.

Operating engineers may be interested in knowing a little of the inside dope regarding the type of research that preceded the building of modern marine power plants. Basically, engineering progress can only go ahead within the limitations of the metallurgical arts. In the early days when puddled wrought iron was the best metal available, and scotch boilers the only type of marine boiler in use, 100 pounds steam pressure was considered the ultimate. Carbon steels and water tube boilers opened up new possibilities and at the beginning of World War I, 200 pounds pressure and 50 deg. F. superheat was considered quite modern. Steam pressures and temperatures were gradually increased, but even as late as 1925 the best central station practice on shore hovered around the 350 lb. 600 F. mark. The metallurgists got busy again, added molybdenum to their steel, and stopped the metal growth that took place at high temperatures. Exhaustive experimentation and testing procedures were set up and, in addition to the creep factor, heat shock and many other characteristics of metals were studied. Today, metals are available which will operate continuously at temperatures of 1050 F. and pressures of 2000 lb.

The mathematicians, on the other hand, have calculated feasible steam pressures up to 5500 lbs. at 1600 F.

When one looks back and compares the steam conditions of the new tankers with those of the T2 tankers they may appear high, but looking the other way, at modern central station practice, they appear only moderate.

Except for the difference in the type of metal used in the manufacture of the boilers, piping, valves and turbines, it doesn't make a great deal of difference, as far as safety is concerned, whether the operating engineer is playing around with 450g-750 F. steam or 850g-850 F. steam. It does make a difference, however, in the corrosion factor in his boilers, and he wants to make sure that he knows all about feedwater de-aeration before he tackles the job. Corrosion takes place rapidly at high temperatures if the free oxygen is present in the feedwater.

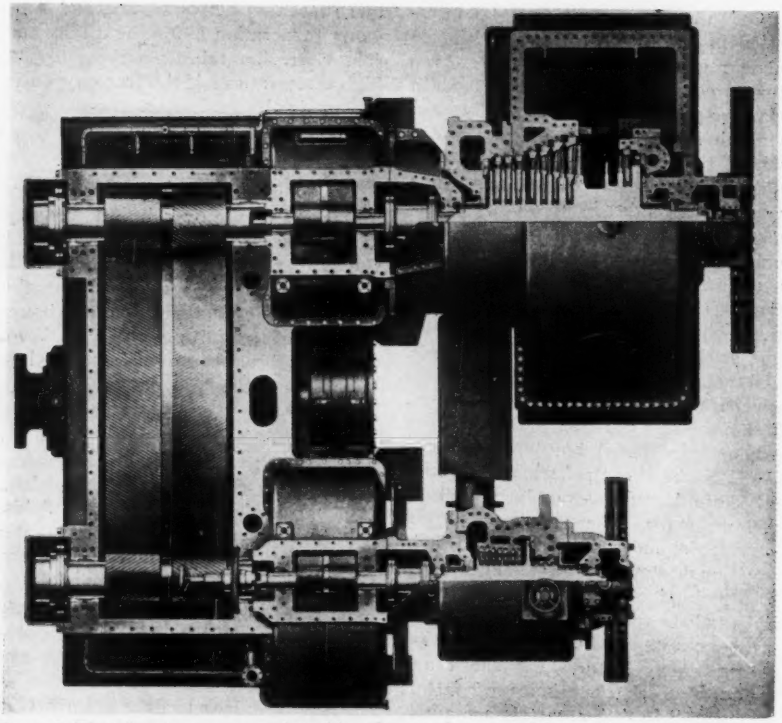
The General Idea Underlying Modern Steam Power Plant Design.

Briefly stated, there are three ways of increasing the overall fuel economy of a steam power plant; one, generate the maximum amount of steam with the minimum fuel; two, convert the heat energy in the steam into maximum amount of useful power; and three, return the maximum amount of low head heat to the system.

All of the foregoing elements are closely co-related and interdependent, and in the new tankers provision has been made for obtaining the best economy in the three respective fields. The boilers are fitted with automatic combustion controls and measures taken for maintaining low stack temperatures.

The main turbines, and turbines driving the auxiliary generators, operate at high initial steam pressures and temperatures and are the most efficient of their type. Hand valves are provided on the propulsion turbines to minimize throttling losses. Four extraction openings are provided in the turbine, which furnish steam for heating the feedwater in progressive steps. Low pressure steam is also available from the same openings for use in the first stage coils of evaporators, and for heating purposes. The extraction feature reduces the steam entering the condenser to a minimum, and boosts the feed-water temperature to a maximum. Drain coolers are also provided for salvaging the heat contained in the miscellaneous drips.

From the foregoing arrangement it is easy to visualize that the main turbines are the heart of the power system. In one way, they are very similar to turbines built for industrial plants requiring large amounts of process steam.



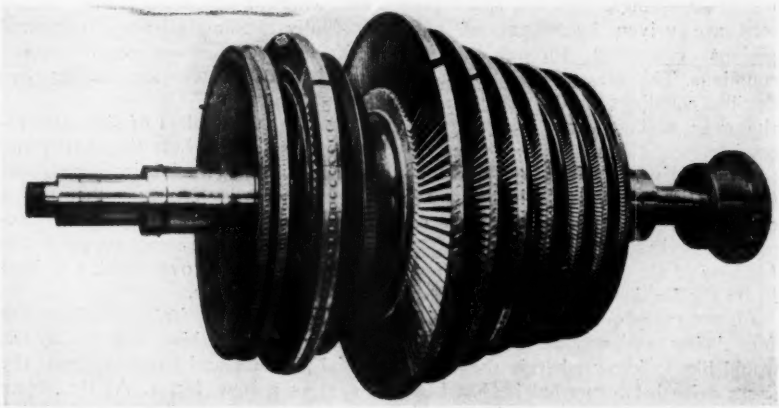
General Electric cross-compound turbine for marine propulsion with double-reduction gear unit. Wash-drawing of top view with parts in semisection. This is typical of those being installed in new super tankers.

Description of Propulsion Turbine Gear Units.

The propulsion units being furnished by the General Electric Co. for a number of the new tankers, consist of a cross-compound, impulse type of steam turbine, and a double reduction gear with the main thrust bearing incorporated in the forward end of the gear casing. The unit is compact and so arranged that the condenser can be underslung directly beneath the turbine and be bolted directly to the exhaust casing of the low pressure element with metal-to-metal contact

The steam conditions at the superheater outlet of the boiler, are 850-lb.g.850 F. Allowing for a nominal pressure and temperature drop between the boiler and turbine it is expected that a steam pressure of 835 lb.g. and temperature of 840 F. can be consistently maintained at the turbine throttle. The turbines are designed for a back pressure of 1.5 in.Hg. absolute or 28.5 in. of vacuum referred to a 30 inch barometer.

The units are designed to deliver 12,500 Shp. at 112 Rpm. under normal operating conditions, and 13,750 Shp. at 115.7 Rpm. under maximum conditions. Under light draft operating conditions the unit is capable of operating continuously at maximum Shp. providing the propeller does not exceed 123 Rpm.



Rotor for low pressure element of G. E. cross-compound geared marine steam turbine. Reversing stages in foreground.

Ample provision also has been made in the turbine design for fluctuations in steam pressures and temperatures. The maximum continuous pressure is limited to 885 lb.g and on momentary swings to 960 lb.g. Temperature is more of a limiting feature in turbine design than pressure, as a continuation of high temperature affects the life of a turbine. For this reason a time limit is placed on high temperature operation. For the units under consideration a temperature limitation of 875 F. has been set for operating periods aggregating not more than five per cent of the time, and at 900 F for periods aggregating not more than one per cent of the time.

The astern turbine is incorporated in the low pressure casing of the turbine. It will develop 80 per cent of rated ahead torque at 50 per cent of rated ahead Rpm.

The turbines are also arranged for emergency operation should anything happen to either the high-pressure or low-pressure element. Flanged openings, blank flanges, filler rings, and orifices are provided for making the temporary steam connections.

The Reduction Gear.

One of the most interesting phases of modern marine engineering progress is the successful development of double reduction gearing. Today gears of this type are available in all powers up to the limitations set by propeller design. Machine tools have been developed for hobbing helical-type teeth on gears up to 200 inches in diameter. To maintain the accuracy required for successful operation the machining is carried out in buildings where the temperatures are accurately controlled.

The meshing arrangement of the reduction gears being furnished for the new tankers by the General Electric Co. is known as the two pinion-articulated type The gear tooth form is the double helix or herringbone type. The articulated type of gear is very flexible as the two high speed pinions each drive their own train of high speed gears without a locking arrangement between the two high-speed elements.

The gear casings are fabricated out of heavy steel plate, welded, and then annealed to relieve all internal stresses. Flexibility between the turbines and the high speed pinions is provided by

flexible couplings. The low-speed pinions are driven by means of quill pinions connected through flexible couplings. This arrangement places the flexible couplings at the after end of the gear casing where they are accessible.

All rotating elements are dynamically balanced, and tooth contours are checked by scientific instruments especially designed for the purpose.

Looking at the Problem of Operation From the Engineer's Point of View.

Almost every operating engineer, before being assigned to a new ship, would like to know wherein the differences arise between what he has been operating in the past, and what is aboard his new ship.

Looking first at the subject of power rating, there is no fundamental difference in the operating procedure on a 12,500 Shp. turbine-gear unit, and one rated 6000 Shp. Both units must first be operated with a turning gear during the warming-up period, and heated gradually to allow all parts to expand uniformly. All turbines require proper draining of condensate during this period.

Turbines, regardless of size, also require gland seals where the shaft penetrates the casing; a lubricating oil system; a steam governing system and a protective device system which effectively closes off the steam supply to the turbine in case of overspeed, and low-pressure oil supply.

Eliminating the features which are common to all turbines leaves only the initial pressure and temperature of the steam as a new factor. As the steam progresses through a turbine the pressure and temperature of the steam drop very rapidly. By the time the steam has passed through the first stage nozzles, there is no foundamental difference between a turbine built for high pressure and one built for moderate pressure.

(Part Two—February Issue).

AUXILIARY POWER PLANTS ON THE NEW SUPER TANKERS.

If one had made the prediction 30 years ago that turbines for auxiliary generator drives aboard ship would some day be operating at 10,000 Rpm., he would have been accused of being a day dreamer.

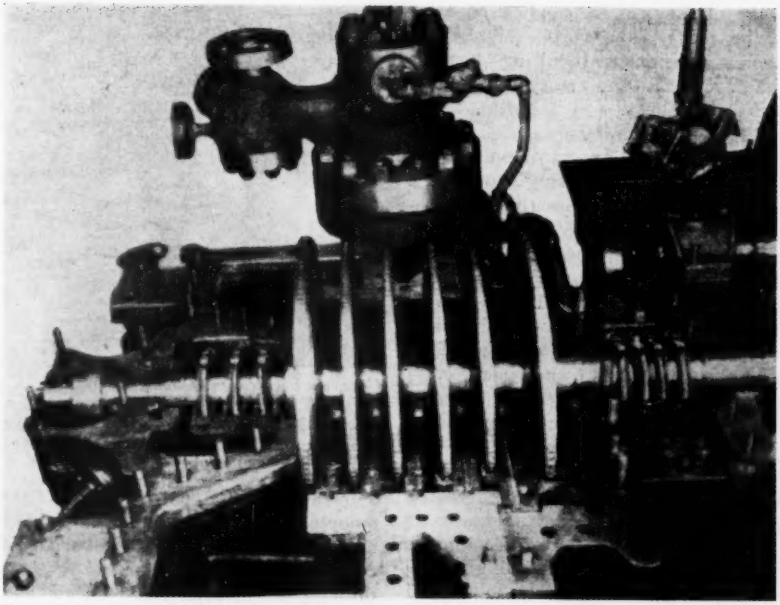
The auxiliary turbine-generators being furnished by the General Electric Co. for a number of the new class of super tankers, however, are being driven by steam turbines having a speed of 10,059 Rpm.

Size is no criterion of power today. Irrespective of the miniature size of the sets, they are designed to generate 400 Kw. of power day in and day out. On occasions, they can generate 500 Kw. of power for a two-hour period just to show that they are not mere babes in the woods.

These sets generate 60-cycle current at 450 volts. The current can be used at this voltage or stepped down by means of transformers to 220 or 110 volts. Lighting circuits and small fractional horsepower motors prefer the latter volage.

Two of these, turbine-driven auxiliary a-c generating sets are to be installed on each super tanker.

The steam conditions are to be the same as those of the propulsion tur-



General Electric 400 KW Geared D. C. Steam Turbine-Generator set for marine service, Navy type, without turbine casings. Close-up right-side view from above.

bines: i.e.—835 PSIG, 840F. at the throttle, and 28.5 inches of vacuum at exhaust.

Considering the fact that the two sets may be operated in parallel, this gives the new ships an abundance of electric auxiliary power.

The General Electric Co. sets deliver a Kw hour to the generator terminals for only a fraction over 11 pounds of steam. Interpreted into terms of "pounds of steam per horsepower hour" this amounts to about 8.3 pounds. In other words, one gallon of water boiled and superheated, then expanded through the turbine creates enough power to cover all losses, and still give a net output of 1,980,000 foot-pounds of energy in usable electric form at the input terminals of the auxiliary motor drives. This will prove to a lot of engineers that steam is a long way from being a dead issue in the new order of things.

The high efficiency of the new sets indicates that all of the electric auxiliary power for the new ships can be supplied for about five per cent of the total steam flow from the boilers. Originally the design of these high-speed sets was for special application on naval vessels. Their use on merchant ships depends upon the length of the trade route, the importance of economy, and the importance of weight and space reduction.

There is only one question that the operating engineers may wish to ask, and that is: "Why was a-c selected in preference to d-c?" This is an easy one to answer. Alternating current motors are tough, contain neither commutator nor brushes, do not spark, and have very low maintenance costs. It is the modern way of doing things—if you can get away with it. On a tanker especially, there are no special motor applications that can be the tail that

wags the dog. If the application were to be made on a cargo ship having about 20 winches requiring dynamic braking, well—the story might be different.

Description of Sets.

Each auxiliary generator set consists of a 6-stage condensing turbine, a single-reduction gear with speed reduction ratio of 10,059 to 1200 Rpm, a 400 Kw, a-c, 0.8 pf, 450-volt, 60-cycle, 3-phase generator, and a shunt-wound exciter generator rated 125 volts, d-c, 1200 Rpm, all coupled together and mounted on a common bedplate. Each set exhausts into its own condenser.

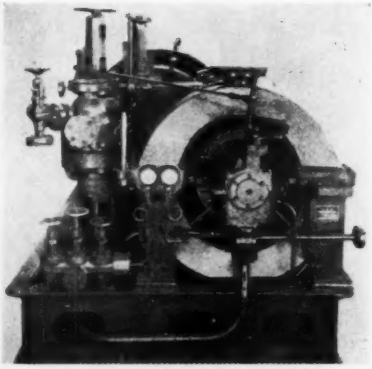
All appurtenances necessary for complete operation are mounted on the set. These include such items as steam seal control valves, oil strainers, oil coolers, steam control valves, and governing system. The lubricating oil tank is situated in the base. Its oil pump is driven by means of a spiral gear attached to the shaft of the low speed gear. Oil level indicators, are handily located for visual inspection. Each set is also supplied with a sensitive and adjustable speed control device for use when paralleling.

The exhaust end of the turbine is carried from the base on vertical supports which are rigid in a cross-axis direction, but are flexible in an axial direction. This allows for axial expansion of the turbine casing under load conditions.

The turbine shaft and gear pinion are rigidly connected through a solid coupling and are supported by three bearings, two in the gear casing and one on the exhaust end of the turbine. A thrust bearing situated at the outboard end of the pinion maintains the axial alignment of the turbine rotor.

Steam under high pressure and high temperature conditions enters the turbine through a strainer, thence through a throttle valve, and control valve to the first stage nozzles.

The speed of the unit is controlled



General Electric 400 KW geared D. C. Steam Turbine-Generator set for marine service, Navy type, end view. These are similar to those being built for the super-tankers.

automatically by a centrifugal governor mounted on top of the oil pump drive shaft. The action of the governor is transmitted to the control valves through a hydraulic relay system and linkage arrangement.

The throttle valve is provided with both a hand wheel for manual control and an emergency tripping device. This valve also trips automatically in case of overspeed.

The low-speed rotating element of the set consists of the reduction gear with its shaft rigidly coupled to the shaft of the a-c generator. It is supported by three shaft bearings, two of which are located in the gear housing and the other on a pedestal located at the exciter end of the a-c generator. The rotor of the exciter generator is mounted on an extension of the a-c generator shaft outside of the bearing pedestal.

Lubricating Oil System and Bearings.

The lubricating oil system performs two functions; one, furnishing oil for lubricating the turbine, gear and generator bearings; and two, furnishing oil to the hydraulic system of the governing mechanism. By coupling the two systems from a single source the

steam control valves will automatically close in case of loss of oil supply. A separate, hand-operated, oil pump is mounted on the gear casing to supply oil to the hydraulic mechanism and bearings when starting the turbine from a standstill.

The bearings are babbitt lined and split horizontally to allow ready disassembly for inspection.

Description of the Turbine.

The bucket wheels, shaft, coupling flange, and balancing rings are integral, being machined from a solid alloy steel forging. After the buckets are properly assembled on the wheels, and the shroud bands riveted in place, the completed rotor is dynamically balanced.

The turbine casing is split horizontally between the high-pressure head and the low-pressure casing, and also horizontally on the centerline. All joints are bolted metal-to-metal.

The steel diaphragms which carry the nozzle partitions are split into upper half and lower half sections. The inner side of the turbine casing is recessed for receiving the diaphragm assemblies. The inner bores of the diaphragms are fitted with metallic packings for restricting the leakage of steam along the shaft.

The first stage nozzle plate is made of corrosion-resisting steel and contains a number of nozzle groups, each connected to an inlet port on the high pressure side of the plate. Each inlet port receives steam through a seamless steel tube which connects it with the governing valve steam chest.

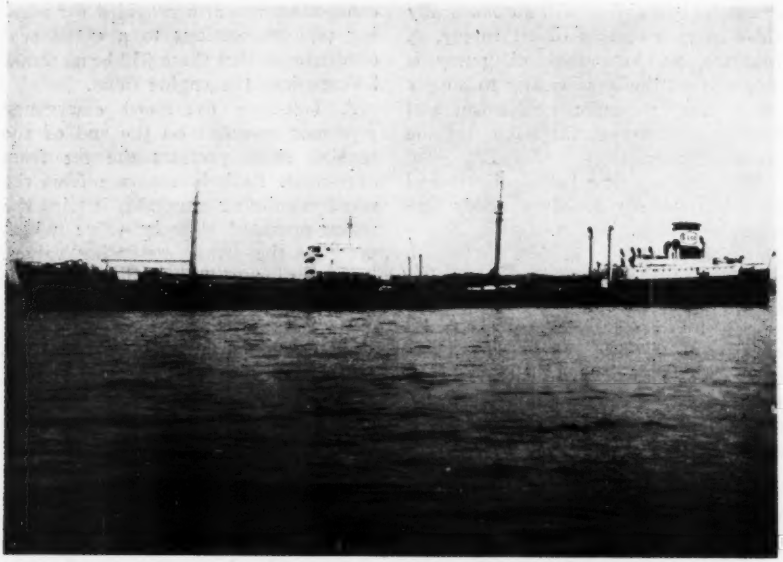
Shaft packings are provided where the shaft projects through the two ends of the casing. They consist of self-lubricating carbon rings split into segments. Each circular group of segments is held in place by garter springs, and is mounted in a packing box. Provision is made for admitting steam to the shaft packing boxes to seal the shaft against air leakage into the turbine. Proper connections are also provided for making pipe connections to a gland seal condenser so that there will be no steam leakage into the engine room.

A bolt-type overspeed emergency governor mounted on the end of the turbine shaft protects the set from overspeed. Entirely separate from the speed regulating governor, it trips the spring-operated throttle valve which precedes the steam governing valves, in case the speed exceeds normal by more than 10 per cent. It is purely a protective device.

Speed Governing During Normal Operation.

Alternating-current generators operating at constant frequency require close speed regulation from no load to full load. This is accomplished by means of a speed-sensitive element of the fly-ball type, and a pilot valve which admits or drains oil from the cylinder of a hydraulic-type of steam governing valve. This in turn opens or closes the ports which admit steam to the first stage nozzles. A micrometer adjustment is supplied on the pilot valve to change the relative position of the oil ports when synchronizing the set with another machine or when dividing the load between two machines operating in parallel.

The speed governor, mounted on top of the reduction gear casing, is driven from an extension of the oil pump shaft. The fly-ball weights of the governor are forced out by centrifugal action and are opposed by a spring under tension. The piston of the pilot valve, connected to the governor by means of a flexible spring coupling, moves up or down with any change in the position of the fly-balls. The cylinder surrounding the piston contains the oil ports which either admit oil to, or drain oil from, the hydraulic cylinder. The relative position of this cylinder with respect to the piston is altered by means of a hand wheel when making manual adjustments. It



The turbine driven tanker "Esso Bethlehem," arriving at New York from Aruba.

is also provided with a small motor which can be remotely controlled from the switchboard.

The spring-loaded piston in the operating cylinder of the steam governing valve is opposed by the pressure of the oil delivered to it by the pilot valve. Stability for any operating condition is definitely obtained by a "follow-up" or restoring mechanism.

When the oil pump pressure becomes too low, the spring in the operating cylinder automatically forces the governing valves closed, thereby protecting the set against low oil pressure.

There are six poppet-type governing valves which admit steam to the first stage nozzles. The valve stems of varying length are so arranged that they operate in pairs. They are raised or lowered by means of a lifting disc responsive to the action of the piston in the hydraulic cylinder.

The foregoing arrangement permits the governor to control automatically the steam supply over the entire operating range. This reduces the throttling losses to a minimum and gives best economy at all loads.

Reduction Gear.

The reduction gear is of the single-helical, single reduction, side-offset type. The pinion is a solid carbon-steel forging, integral with its shaft, and is heat treated for long life. The low speed gear wheel and gear shaft are made from carbon-steel, heat-treated forgings. The couplings for both elements are of the solid type.

The steel gear casing is split at the horizontal centerline of the gears into an upper half and a lower half section. The upper half serves as a bearing cap for the gear and pinion bearings and as a cover for the gear unit. It also serves as a support for the pilot valve bushing, its housing, and the speed changer. All joints in the casing are bolted metal-to-metal and are oil-tight.

Three openings with hinged covers are provided in the upper half of the

casing for the inspection of the speed governor gear teeth, and balancing plugs in the coupling flange of the turbine shaft.

Description of A-C Generator.

The generator is of the salient pole, open drip-proof, self-ventilated type, rated 400 Kw. at .8 pf continuous, and 500 Kw. at .8 pf for 2 hours. For five minutes, 150 per cent current at .5 pf at 450 volts can be maintained. It is designed for operation in an ambient temperature of 50C. Insulation is type A in the stator, and type B in the rotor.

The allowable temperature rise under normal continuous load is 40C. by thermometer on the stator, and 70C. by resistance on the rotor. Under the two hour maximum overload condition the permissible temperature rise on the stator is 55C. and on the rotor 85C.

The rated voltage is 450. The rated speed is 1200 Rpm, and the rated frequency, 60 cycles per second. There are six field poles on the rotor. Stator winding is of the Y type.

The generator shaft is made from a carbon steel heat-treated forging with the half coupling, for connecting to the gear shaft, made integral.

The rotor consists of a spider constructed out of laminations riveted together and shrunk on the shaft, and the field poles which are made up of punchings riveted together. The assembled pole pieces are keyed to dovetail slots in the spider. The field coils are series-connected with their terminals connected to two collector rings. These rings are mounted on a steel sleeve pressed on to the shaft.

The stator frame is made of steel plate, while the stator core is made of high-grade magnetic steel punchings. Stator windings are of the formed type. The a-c leads are brought out as a set of six bus bars beneath the generator.

The generator is also provided with strip heaters for prevention of condensation on the windings when the set is not in use.

Description of Exciter Generator.

Excitation for the field windings of the main generator is supplied by an exciter generator whose rotor is directly connected and overhung from the end of the main generator shaft. The frame of the exciter generator is supported from the base on feet.

The exciter is shunt wound, rated 7.5 Kw. continuous at 125 volts d-c, 40C. rise. It is of the open drip-proof, self-ventilated type.

Description of the Voltage Regulator.

The voltage regulator (a switchboard device) maintains constant voltage at the terminals of the a-c generator by controlling the field of the exciter set. This in turn supplies current to the main generator field poles.

The voltage regulator consists of two elements: one, a voltage-sensitive element, and two, a regulating resistor.

The coil of the voltage-sensitive element is excited by direct current, derived from the system voltage through potential transformers, voltage adjusting rheostat, fixed resistors, and rectifiers. The voltage applied to the coil is the average of the three phase voltages.

The regulating resistor has 54 taps connected to 54 flexible fingers which rest on a shorting bar. The fingers are lifted off of the shorting bar successively by means of two insulated finger lifters which are attached to a spring-restrained armature. The coil magnet core which reflects the line voltage acts magnetically to lower or raise the armature as the occasion demands.

The resistor element is connected in series with the field of the exciter generator which furnishes the current for exciting the field of the main generator. A stabilizing transformer is used to prevent overshooting and hunting.

A voltage adjusting rheostat is inserted in one of the a-c leads to the coil of the voltage-sensitive element, so that the operator can make any small adjustment required to set initially the regulator for the exact voltage desired.

There is also a cutout switch on the unit which has three positions: Manual, Test, and Automatic. In the manual position, the regulator is cut out entirely from the field. In this case the voltage is controlled by a manually-operated rheostat. In the test position, the coil of the regulator is energized with the resistor short circuited. In the automatic position, the entire unit functions as previously described. The regulator is designed to hold the main line voltage within plus or minus one per cent from no load to full load with the equalizer reactor short circuited.

Proper division of the reactive kva during parallel operation is accomplished by means of a current transformer and an equalizing reactor.

RIVETED VS. WELDED SHIP STRUCTURE.

ACKNOWLEDGMENT.

This article, by MR. E. M. MACCUTCHEON of the David Taylor Model Basin, is reprinted from "The Welding Journal" for February, 1949. It discusses the relative merits of riveting and welding with regard to the incidence of cracks. Because of the importance of the subject discussion has also been reprinted.

ABSTRACT.

Investigations initiated in connection with the cracking of merchant ships during World War II have uncovered new evidence bearing on the relative incidence of cracks occurring in ship structures fabricated by riveting and by welding. These new findings are critically reviewed along with certain widely accepted viewpoints. Available data are compared to test the validity of evidence bearing on the incidence of cracks. It is concluded that there is no evidence indicating that either riveted or welded structure is more susceptible to the inception of cracks.

Introduction.

Riveting had been the principal method of structural fabrication for many years before welding became a serious competitor. The introduction of welding was hampered by the natural inertia which accompanies the replacement of a universally accepted engineering method. Furthermore, it was beset by technical difficulties which made it a good target for criticism and the brunt of many unfair comparisons. The comparative merit of riveting and welding was the subject of prolonged speculation and controversy. Repetition and conjecture wore the subject threadbare. Actually there was such a dearth of facts that many of the original arguments remained unsettled.

Early discussions on this subject were replete with wishful thinking. This was largely engendered by the natural resistance to the birth of a new industry. Welding has now established its place in structural fabrication, and the causes for rancor have been mellowed by time. A reopening of this question should be possible with complete freedom from mercenary bias.

The epidemic of structural failures in merchant vessels has revived many of the original contentions associated with this dormant controversy. In commencing their recent survey of the structural-failure problem, the naval architects were confronted by the same provoking lack of factual data which had spirited much of the earlier debates and which befogged the true nature of the problem. Early recognition of the shortage of facts encouraged the establishment of the broad statistical studies and research investigations which are now bearing fruit in our technical literature. Among the findings of these studies, are many facts relating to the question of riveted vs. welded structure.

The time has come for a reappraisal of the subject. The study of the problem is fraught with practical complications. It is difficult to separate the influencing variables and study them one at a time even in the research

laboratory. Much of the apparent contradiction of earlier tests results from a failure to recognize the importance of certain influential parameters. Recent studies have emphasized notch sensitivity. This means that the toughness of the steel and the temperature of the test are important to the results. Early tests seldom included control or even recognition of these effects. A recent tensile test on a riveted butt joint with a nominal efficiency of 85% resulted in a fracture in the base plate starting at a sheared edge and completely avoiding the riveted joint. This test was made at —20° F. It is clear that the notch sensitivity is important to riveted as well as to welded structure, but the relative influence may not be the same.

A careful statement of the problem is essential to an evaluation of the new evidence bearing on the relative merits of fabrication by riveting the welding. There are many bases for making the comparison between riveting and welding, for example, construction costs, maintenance costs, weight economy and structural performance. The following discussion concerns the structural performance only.

The ultimate merit of a structure depends on its ability to resist failure. In a ship's hull, failure may be of two types, a crack or a buckle. The structural performance of the hull may then be measured by three yardsticks: (1) Its ability to prevent the formation of buckles, (2) its ability to resist the propagation of cracks and (3) its ability to resist the starting of cracks.

It has been amply demonstrated that the riveted structure, with faying flanges and stiffening overlaps will present more resistance to compressive loading than its welded counterpart when functioning in a ship's hull. Likewise the evidence, accumulated during the last few years, firmly establishes the crack stopping capabilities of a riveted seam. By providing a sufficient number of carefully located riveted barriers, it would be possible to build a welded ship which is equal to a riveted ship, in so far as resisting the propagation of cracks is concerned. The relative merit of such a structure, with regard to cracking, would then depend entirely on the likelihood of crack inception. This is the third yardstick of structural performance and it is intended to limit the remainder of this paper to discussion of this third yardstick, i.e., the ability of the structure to resist the starting of cracks.

It was the intention of the author to assemble some of the more recent data bearing on the incidence of cracking and to test their validity in this paper. By accumulating all the resulting findings in a logical array it was supposed that new facts would make themselves evident. The course of this paper was set toward a noble destination; but the reader is hereby warned to curb his expectations lest the following discussion appear anticlimactic. Practically all of the findings are negative. Negative findings are important, however, even though they may be much less interesting than positive results. For the case in question, the negative findings take on additional importance because they contradict widely voiced opinions upon which engineering judgments are based every day. The opportunity to contend such viewpoints supplied the incentive to complete the discussion which follows.

Evidence bearing on the resistance of the structure to the starting of cracks can be more easily handled if it is divided into two groups; first, direct evidence statistically relating the occurrence of cracking with the method of fabrication (riveting or welding), and second, indirect evidence. Indirect evidence includes data by which the method of fabrication can be shown to influence factors which, in turn, influence the cracking.

Such factors must, of course, include only those which are known to have a real influence on the incidence of cracking.

The voluminous study of structural failures during World War II provided ample opportunity to survey the fractures in welded ships and become familiar with their detailed characteristics. A similarly detailed study has been accomplished on at least one riveted ship, the *Oakey L. Alexander*. This latter survey shows that features of the failure were similar to those observed on the welded ships. The similarity of the fractures is important. If the fractures were not similar, it is probable that their occurrence would be encouraged by a different set of influencing factors. The fact that the fractures are similar admits the ready and direct comparison of data bearing on the influencing factors.

The important factors, governing the incidence of cracking, are type of steel, state of stress (constraint), temperature and magnitude or rate of loading applied to the critical region. Indirect evidence will hardly ever be quantitative in nature. The causes are twice removed from the effects, and a quantitative relation between the influencing parameters and the likelihood of cracking has been established in only a few special cases.

Presentation of Evidence Relative to the Incidence of Cracking.

For ships' hulls constructed by riveting and by welding the following direct evidence has been submitted as indicating superiority of riveting with regard to the incidence of cracking: (1) The relative number of ships which broke in two when hit by torpedoes, (2) the reputedly higher incidence of fracture in the welded ships and (3) the comparatively few serious failures occurring on the Bethlehem Fairfield Liberty Ships.

As direct evidence indicating superiority of welding the following may be cited: The higher incidence of cracking in the ships fitted with riveted crack arrestors.

Indirect evidence was stated, above, to include data bearing on type of material, degree of constraint, temperature and the magnitude or rate of applied loading. Obviously the method of fabrication cannot be expected to influence the type of material or the temperature. The applied loading refers to the magnitude of the stress values in the parts of the structure where trouble is likely. Locked-in stresses are, therefore, considered a part of the loading. Of the factors controlling fracture, both loading and constraint can be influenced by the method of fabrication (riveting or welding).

Welding involves cavities, underbead cracking, etc. Riveting involves sheared edges and rivet holes. Few qualitative and no quantitative statistical data have been obtained, bearing on the relative danger from notches typical of the two types of fabrication. This means that there is no indirect evidence bearing on the degree of constraint. Likewise, there have been no comparative tests on the two types of structure which involve rate of loading. Hence, all the indirect factors except the magnitude of load, are eliminated from consideration because of the lack of information. Some data have been obtained on the effect of riveting and welding on the magnitude of the stress, i.e., the loading in critical regions, and this is the only indirect evidence which it is possible to assemble.

Indirect evidence indicating lower stress values in the riveted structure includes the following: (1) Numerous reports of higher flexibility and lower stresses in the riveted ships, (2) the construction stresses locked in the decks of the ships with riveted gunwales and a riveted deck seam are reported to be compressive and higher than those recorded in corresponding locations of ships with welded gun-

wales and (3) tests of riveted joints indicate a nonelastic behavior at high stresses.

Some of the indirect evidence may also support the possibility that no difference results from the type of structure. Included are the following: (1) An examination of cap welds spanning riveted seams and in way of welded butts failed to reveal any cracks and (2) as a result of their tests on the two sister tankers, the *Neverita* (welded) and the *Newcombia* (riveted), the British report no substantial difference in the behavior of the two structures, with regard to deflection of the hull.

Discussion of Direct Evidence.

Records are available for study of 331 riveted or welded ships which were torpedoed during World War II. Of these vessels 48 broke in two. For the ships constructed by riveting, those breaking in two made up 14% of the total number struck by torpedoes. For the welded ships, the percentage was 18. The number of ships involved in this study is nowhere near sufficient for a valid statistical conclusion. But, even if the sample were adequate the results would not prove that the incidence of cracking is lower in the riveted ships because the riveted seams prevented the ships from breaking in two in many instances, of which several reported cases can be cited as examples. The evidence is, therefore, invalid, and must be discarded.

Riveted ships have suffered structural distress. The famous liners *Leviathan* and *Majestic* cracked their main decks, and several riveted tankers have broken in two. No longer ago than March 1947, the *Oakey L. Alexander,* a riveted collier 31 years old, broke in two, off the coast of Maine. For four years during World War II a complete central record of structural failures was maintained for all U. S. ships. Great numbers of nearly identical ships were in service as a result of the Maritime Commission construction program. Comparison of the fracture records with the accumulated service data permitted a reasonably accurate statistical study to be made. The large groups of ships, however, were made up almost entirely of Liberty Ships, T2 tankers, Victory ships and other new ships which were practically all welded. The only mass production program of riveted ships occurred during World War I. Few of these ships were still in operation during the period of the structural-failure study described above.

With the exception of the outstanding examples of failure, the structural-performance records of the riveted ships are hopelessly scattered or lost. Many a cracked plate was replaced as a part of the voyage repair of a riveted ship, but we shall never know how many. Here again no conclusion can be drawn as to the relative performance of the two types of fabrication with regard to the incidence of cracks.

The Bethlehem Fairfield Liberty ships had riveted shell seams; the other Liberty ships did not. This inspired the operational managers of the War Shipping Adm. to assign these ships to routes where inclement weather prevailed. The service record of ships built by this yard was above average in so far as the incidence of cracks was concerned. However, it cannot be concluded that this superiority was directly attributable to the presence of the riveted seams. Other factors may have assisted to account for part, or all, of the difference.

The records show that the training and supervision of labor as well as the other fabrication techniques employed in the Bethlehem Fairfield Yard were little different from those in other yards. Considering the leveling effect of the thousands of men working on these jobs all personality factors are barred. It must be concluded that workmanship could not have accounted for

the superiority of the Bethlehem Fairfield ships. On the other hand, steel has been proved to be an important factor in the fracture problem. The steel for the Liberty Ships was generally allocated on a nationwide basis, but it has been found that about two-thirds of the steel for the Bethlehem Fairfield ships came from the nearby Bethlehem-Sparrows Point rolling mill. It is not known whether this steel was better than average, but its preponderant use unsettles direct reasoning.

The degree of superiority of these ships over other Liberty ships is not overwhelming. They incurred only 15% as many serious failures as the average, but the percentage is about 55 when all magnitudes of damage are included. One other yard which built more all-welded vessels actually had a better record for all classes of failures. After considering all these factors it must be admitted that the record of the Bethlehem Fairfield ships does not determine, beyond dispute, the beneficial attributes of riveting the side shell as a crack inhibitor.

Evidence which appears to support superiority on the part of the welded structure, must be similarly evaluated. One group of Liberty ships, all of which were completed before Feb. 1, 1943, was submitted to a statistical appraisal. This group included 667 vessels on which the construction practices were of about the same quality and on which the design details were practically identical. There was only one season, i.e., the winter of 1944-45, during which this group of ships was reasonably well-divided between those fitted with riveted crack arrestors and those which had not yet been altered to incorporate the arrestors. During this winter season the ships without crack arrestors accumulated 1620 ship-months of service time and incurred 10 fractures. The ships with crack arrestors accumulated 847 ship-months of service time and incurred 16 fractures. The incidence of fracture for the ships without crack arrestors was 0.006 fractures per ship-month, and the incidence for ships fitted with crack arrestors was 0.019 fractures per ship-month. This is a small sample of data for a statistical comparison. In addition, it is known that the service conditions for the two groups of ships were not equal. The assignment of operating routes depended on the availability of ships which had been fitted with crack arrestors. Such ships were always assigned to routes where severe weather conditions were anticipated. When this fact is considered, it is necessary to reject the study of ships with crack arrestors as evidence bearing on the risk of incurring cracks in riveted or welded structures.

Discussion of Indirect Evidence.

During World War II there was a widespread impression in marine circles that a riveted ship was less likely to incur cracks. It has already been pointed out that there are no statistical data confirming such a rumor. The rumor in question did not originate from the performance of the Bethlehem Fairfield Liberty Ships for the simple reason that, when the rumor was at its height, the Bethlehem Fairfield Yard was just getting into production and the apparently good service record of the ships had not yet been established. Actually, it is believed that this confidence in the riveted structure stemmed from its crack-stopping ability rather than from its ability to eliminate the starting of cracks. The ship masters developed their individual hypotheses for the rumored superiority of the riveted ships. The most widespread explanation was "flexibility." Reports of additional flexibility in riveted hulls were numerous. Efforts have been made to locate a factual basis for these reports, but none has been found. No actual measurements have been taken except in the formal structural studies

which are already receiving much attention. It is believed that the flexibility reports must be disregarded because they completely lack factual support.

It has been reported that the locked-in stresses in a fore-and-aft direction in the decks of welded ships are generally compressive in nature, with values ranging from 8800 psi. in tension to 16,600 psi. in compression. One study showed that vessels with riveted gunwale angles and a riveted deck seam, with a seam strap, had higher compressive stresses than a similar group with welded gunwales. There are no test data to dispute this finding. It should be studied from a statistical viewpoint, however. Only six ships with welded decks and five with riveted gunwales and deck seams are included in this study. The difference in stresses, about 4000 psi., is small when compared with the scatter of results (10,000 psi). All the 11 ships included in this comparison had riveted shell seams. On the basis of the same tests, the investigators point out that the results from either the group with the welded gunwale, or those with the riveted gunwale and deck, compare well with two other surveys of locked-in stresses. These other surveys included tests on 23 Liberty ships and 26 Victory ships. The hulls of these ships were all welded except that the Victory ships had riveted gunwales. If it is remembered that all 11 ships in the above-mentioned comparison had riveted side shell seams, this evidence cannot be taken to indicate that the presence of riveting influenced the stress locked in the Liberty ship decks.

The fact that riveted joints under test manifest slip in the form of nonelastic behavior cannot be disputed. It is true that, with special techniques of riveting, special materials and careful alignment of rivet holes, the extent of this nonelastic performance can be minimized. Such care cannot be expected in the construction of a ship, so the question reduces to one of whether this nonelastic performance exhibits itself in an average structure under the influence of normal operating loads.

Extensometers placed on a riveted deck butt and adjacent seam during the launching of a new heavy cruiser failed to indicate evidence of slip. The stress in the deck reached 16,000 psi. Due to the fact that the riveted butts in a ship's hull are staggered, a certain amount of the longitudinal load is taken by the rivets in the seams between the staggered butts. The proportion of load carried by the seams varies from nearly zero at low loads to as much as 45% when failure is imminent. The longitudinal load in the seams is mainly controlled by the eccentricity of the butts at low loads and rivet slip as the load increases. In the case of the above-mentioned cruiser it is known that the area of riveted cross section in shear in the butts is about equal to the corresponding section of deck plating. Hence, it can be said that the stress on the rivets during this test had a maximum value of 9000 to 15,000 psi. The spread of maximum values is due, of course, to uncertainty of the load on the seams.

Measurements on the destroyers *Preston* and *Bruce* did not include direct measurements of rivet slip, but some of the gages were placed across deck butts. In this test the stress values for which no evidences of slip were recorded were 6000 to 12,000 psi. At higher loadings the elongations were not recoverable and leakage occurred in underwater butts. Slip had certainly occurred. The fact that these ships, two old and one new, showed no slip, until the above-mentioned stress values were reached, is interesting. Laboratory tests of typical ship construction joints confirm these results with reports of slip for values of 6000 to 7000 psi. on the rivets.

Service records show that hundreds

of loose rivets must be replaced in a ship which has weathered a particularly heavy storm. This is a common occurrence, and it must be concluded that rivets do slip under severe operational conditions. On the other hand, the tests show that it must take moderate stresses to promote slip (above 6000 psi. on the net section of the rivet).

In ships constructed with welded butts and riveted seams it is common practice to fuse one strake of plating to the next by means of a short fillet weld deposited in way of each welded butt. These welds are usually about 3 in. long and are commonly known as cap welds. Before an appreciable shear deflection could occur in the riveted seam, it was supposed that such small welds would show evidences of distress, either in the form of yielding or cracks. An examination of these welds on several Liberty ships, which had already seen extensive service, failed to reveal any cracked cap welds. Even when visual inspection was supplemented by magniflux it was not possible to detect cracks. In order to verify this finding a test was made of a large specimen fabricated from 1- and ¾-in. plating and including typical cap welds in shear. A shear displacement of nearly 1/32 in. occurred before the cap weld cracked. If rivet slip occurred in these ships, the magnitude of slip was evidently less than 1/32 in.

The remaining indirect evidence comes from the British test program on the *Neverita* and *Newcombia*. The report on the *Newcombia* includes a conclusion with regard to the comparison between these two ships which reads as follows: "The stress distribution and longitudinal deflections in the two forms of construction show no major differences." This conclusion should be interpreted in terms of flexibility and slip stress with the greatest caution. The theoretical deflection of

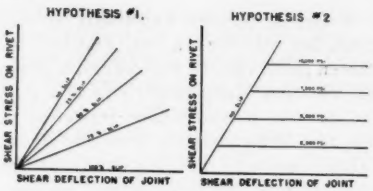


Fig. 1

the hull resulting from rivet slip depends on the particular hypothesis for rivet slip assumed. In order to demonstrate this, calculations were made on the *Newcombia* in hogged condition to show the amount of hull deflection which could be expected as a result of assuming two separate hypotheses for rivet slip.

Under Hypothesis 1 it is assumed that the stress on each rivet is a definite fraction of the stress, which would have been on the rivet, had no slip occurred. Hence, for any given degree of slip the shear-deflection for the various joints is proportional to the applied stress. Under Hypothesis 2 it is assumed that slip is nonexistent until a certain stress level is reached, in the rivet. It is assumed that the rivet stress cannot increase above this level, hence, further increases in shear deflection occur at a constant maximum stress value. The two hypotheses are shown graphically in Fig. 1.

Figure 2 shows the deflection of the ship's hull corresponding to various amounts of rivet slip for each of the two hypotheses. A careful study of the test results indicates that the deflection of the riveted ship was actually 4 to 7% higher than that of the welded ship. If Hypothesis 1 is accepted, this additional deflection would correspond to a very small percentage of rivet slip, Fig. 2. On the other hand, if Hypothesis 2 is considered to be the acceptable explanation, a low value of slip stress must be assumed in order to correspond to the slight additional deflection. The slip stress would be only about 5000 psi. for 4% additional deflection.

Probably neither hypothesis is correct, but the true action may lie between these two. If slip occurs, a stress distribution study might reveal a clew as to the proportionate influence of the two hypotheses for rivet-slip stress. This requires a detailed study of the test results with extensive calculations. Work along these lines has been started but is not completed. One fact is clear at this state: The presence of the welded longitudinal bulkheads in the *Newcombia* masks the difference in performance between it and the all welded *Neverita*. A greater contrast will undoubtedly result from the comparsion between the *Ocean Vulcan* and the *Clan Alpine;* these ships have no longitudinal bulkheads.

The bending loads on the hull of a ship are controlled by the longitudinal distribution of weights and buoyancy and by certain dynamic and form characteristics which govern the ship's reactions in a seaway. None of these factors are influenced by the type of fabrication used in the hull. The internal resisting moment in the hull structure must be in harmony with external conditions at all times. This is an important fact, which is frequently overlooked when glorifying the stress-ameliorating benefits to be expected from rivet slip. Slip in the riveted joints can redistribute the stress in

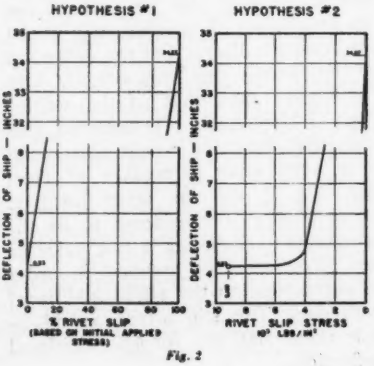

HYPOTHESIS #1
HYPOTHESIS #2
DEFLECTION OF SHIP — INCHES
% RIVET SLIP (BASED ON INITIAL APPLIED STRESS)
RIVET SLIP STRESS 10³ LBS/IN²

Fig. 2

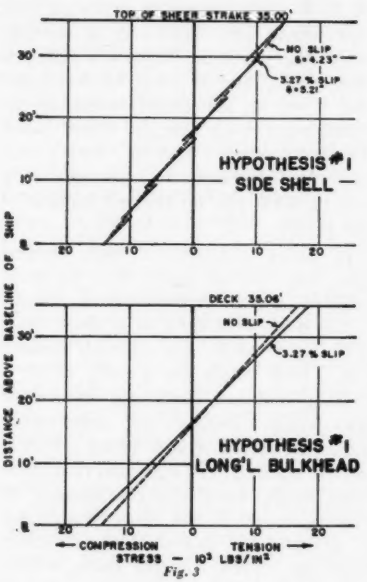


Fig. 3

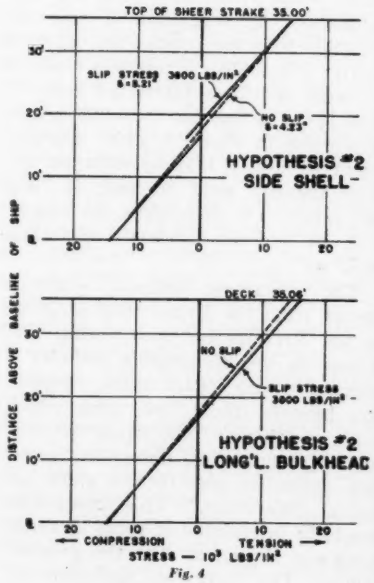


Fig. 4

the structure, but the internal bending moment cannot be changed. Figures 3 and 4 show the internal stress distribution amidships necessary to meet the requirements of external loading. Figure 3 indicates the stress distribution corresponding to Hypothesis 1, which was previously described. Figure 4 shows the distribution for Hypothesis 2. Under either hypothesis, it can be seen that, while some portions of the structure incur stress reductions, other portions suffer increases. It is clear that the effects of rivet slip on stress magnitude are more likely to be harmful than to be beneficial because the peak stresses become higher and higher as the structure subdivides into less competent units. This would be particularly serious if the peak stresses happen to occur at critical locations.

General Discussion.

Other suggestions have been proposed regarding the possible influence of riveting and welding on the stress distribution in the ship hull. Riveted joints involving overlaps naturally produce a load-elongation relationship differing from that of a steel plate because of the eccentricity of the joint. The longitudinal seams of a ship always consist of eccentric overlaps of this type. It has been maintained that the presence of these seams is influential in reducing the transverse stresses in the deck and bottom of the ship thereby decreasing the degree of biaxiality and perhaps the incidence of cracks. An examination of the transverse stresses in the deck and bottom of the *Neverita* and *Newcombia* fails to reveal a pronounced difference in magnitude. Although the deck-stress range is less on the *Newcombia* (riveted), the scatter of values seems to have been influenced considerably more by local effects on both ships than by the presence of the riveted seams. These ships are tankers, with longitudinal framing in deck and bottom. The transverse stresses are lower than on transversely framed ships. It is possible that parallel tests now underway on the transversely framed *Ocean Vulcan* and *Clan Alpine* will show a difference.

Another hypothesis suggests that portions of the structure incurring high locked-in stresses during fabrication adjust themselves because of slip in nearby riveted seams. In this manner stress peaks are reduced with a possible decrease in the incidence of cracking. There are still other plausible proposals, some of which might yield valuable findings if they were studied.

The direct data on the Bethlehem Fairfield ships might be validated by a careful survey of steel distribution and quality. It would be necessary to establish that the quality of this steel was no different from that of the general supply, or that the steel in critical locations came from other mills than Bethlehem-Sparrows Point rolling mill. If it should turn out that better steel was used in the critical locations, the riveted seams could not be given any credit for reducing the susceptibility of the hull to cracking. It is suspected that even after completion of a laborious steel survey, the findings would be statistically weak. There is little hope of validating the other direct evidence.

Tests on the transversely framed vessels *Ocean Vulcan* (welded) and *Clan Alpine* (riveted) will double or triple our present knowledge of the influence of riveting on the stress distribution. There is a need to know more about the relative constraint in riveted and welded joints. Tests of typical riveted joints at low temperatures would be good for comparison with welded design detail tests now under way.

Summary.

A formidable array of objections has been assembled in dispute of each part of the evidence. Summarizing the discussion we find that practically all

evidence must be discarded for one reason or another, only a few shreds remain valid. The findings are summarized below with suitable qualifications.

A. Facts regarding the relation between the method of fabrication and the incidence of cracking.
   1. Both types of structures crack.
   2. The characteristics of the fractures in the two types of structure are similar.

B. Facts regarding the relation between the method of fabrication and physical parameters which are known to influence the incidence of cracking.
   1. Stresses locked in the welded decks of ships are not appreciably affected by the presence of riveted seams in either the deck or side shell.
   2. Riveted joints may slip in service at moderate loading, (above 6000 psi. shear stress on the rivets).
   3. The shear deflection in a riveted seam is usually less than 1/32 in. when it slips in service.
   4. Extensive rivet slip might occur without appreciably increasing the deflection of the hull (see Fig. 2, Hypothesis 2).
   5. Rivet slip under service loads may be accomplished by an increase of stress at critical locations in the structure.

Conclusions.

In considering the following conclusions it should be recalled that the study herein relates to the influence of the fabrication method on the incidence of cracking, and it should be re-emphasized that buckling and resistance to the propagation of cracks are not subjects of discussion.

1. From the evidence examined, it cannot be determined whether a welded or a riveted structure is more susceptible to the inception of cracks.

2. Rivet slip may occur at loads within the normal operating range.

3. Rivet slip could result in an increase of stress in critical portions of the structure.

Discussion by D. P. Brown.

D. P. Brown is Chief Surveyor of the American Bureau of Shipping, New York, N. Y.

In the following comments on this very interesting paper I wish to explain that wherever reference is made to a Liberty ship as being "all-welded" it is intended that reference is made to that type of ship in which the shell, decks and other longitudinal material are completely welded, irrespective of whether the side frames are welded or riveted, as I do not believe that the method of attaching the side frames has any material effect on the problem under discussion. I do not believe that there can be any quarrel with Mr. MacCutcheon's statement that there has been ample experience to demonstrate that the riveted structure has certain advantages in respect to the ability to prevent buckling and the ability to resist the propagation of cracks. I must admit that when it was first decided to introduce some riveted longitudinal joints in the all-welded Liberty ship hulls I was hopeful that the introduction of such joints would not only provide barriers against the propagation of cracks but that these joints would also reduce the incidence of cracking.

In the early stages of the operation of the Liberty ships the incidence of cracks attained a very alarming rate and one which, in my opinion, was considerably beyond that which would

normally be expected based upon experience with riveted structures. I gather that Mr. MacCutcheon may not agree with me entirely in this matter. It is true that cracks were not completely unknown in riveted ships but in most cases these occurred in ships of comparatively old age where deterioration had already become evident or in ships of inadequate scantlings based upon present day standards. Neither of these conditions was present in the Liberty ships and investigation of the cracking appeared to indicate that in the all-welded ships, or in large all-welded areas of otherwise riveted ships, certain details were far more critical than in the completely riveted ship and there appeared to be considerably greater susceptibility to minor defects in workmanship. As a result of these investigations changes were made in the original details such as the introduction of radiused hatch corners, the fitting of detached bulwarks, the serrating of bilge keels and the rounding of scupper and accommodation ladder cuts. Concurrently with these changes a great publicity campaign was instituted emphasizing the necessity for careful workmanship and proper controls of erection and welding procedure.

These measures soon had a very noticeable effect in reducing the incidence of fractures but before a great deal of experience had been gained with the improved structure it was also decided to add the riveted crack arrestors so that the effect, if any, of fitting the riveted joints could not immediately be separated from the effect of the changes in details and improvement in workmanship.

Mr. MacCutcheon has cited some figures for a selected group of 667 vessels operating in the winter of 1944-45 and which indicate that the incidence of fracture in the ships with crack arrestors was more than three times that in the all-welded ships. It is presumed that he has included in the vessels having crack arrestors, those with riveted shell seams and those having only one or two riveted joints in the deck or at the gunwale on each side and that in the fractures counted he has included only those which occurred in the effective strength decks, shell or appendages thereto, within the midship portion of the vessels, excluding the various miscellaneous fractures which may occur in bulkheads and other internal structure or at the extreme ends and which are of the type as could not reasonably be expected to be affected by the riveting of the shell seams or the introduction of a few riveted crack arrestors.

In the American Bureau of Shipping we have been keeping continuous records of all reported failures on those vessels under our surveillance, from the time the first vessel was placed in operation, in an effort to determine the effect of the various modifications, and our records indicate results which are quite at variance with those cited by Mr. MacCutcheon. In our records we have considered the Liberty ships as being of five separate types and for these five types the records of ship-months of operation up to July 1, 1948 and of fractures occurring in the decks, shell or appendages thereto, within the main portion of the hull structure are as follows:

Type	Ship-months of service up to 7/1/48	Number of fractures	Fractures per 1000 ship-months
All-welded with original details	25,200	408	16.2
All-welded with improved details	17,300	52	3.0
All-welded with improved details and crack arrestors	41,200	134	3.3
Riveted seams with original details	4,000	14	3.5
Riveted seams with improved details	13,800	35	2.5

These figures can lead to many interesting speculations but since the point under discussion at the present time has to deal only with the effect of riveting on the incidence of fracture, I will confine my comments to this particular matter. It would appear that the incidence of 3.3 fractures per 1000 ship-months of service for the all-welded vessel with improved details and with crack arrestors as against the incidence of 3.0 fractures for the corresponding vessel without crack arrestors would indicate an adverse effect due to the fitting of crack arrestors but this is by no means anywhere near as great as the more than 3 to 1 ratio indicated by Mr. MacCutcheon's figures. Mr. MacCutcheon has quite properly pointed out that his figures are undoubtedly influenced to a great degree by the policy of selecting those ships with riveted crack arrestors for the more arduous services which were in force during the period covered by his records. This effect of selection will always be present, but to a lesser and lesser degree as the records increase and it is quite possible that this may still account for the apparent unfavorable effect of the riveted crack arrestors even in our over-all figures.

Another interesting fact to be obtained from these records is in respect to the benefits of the improved details. It should be noted that for the all-welded ship with the original details the incidence of fractures is 16.2 per 1000 ship-months and that this figure was reduced to only 3.0 in the corresponding ship with improved details. For the ships with riveted seams the incidence is 3.5 per 1000 ship-months with the original details and it is reduced to 2.5 with the improved details.

As I stated before, there was, concurrently with the program of changing details, an intensive campaign to improve the quality of the workmanship and to establish more rigid controls on shipyard practices and procedures, and to just what extent this campaign contributed to the improvement in the fracture records can never be definitely established. It is not reasonable to believe, however, that the yard building the ships with the riveted seams did not have some room for improvement as the construction program progressed and that it did not also obtain a part of the improvement in the fracture records as a result of this campaign.

Mr. MacCutcheon has suggested that there is a possibility that the steel used in the ships with the riveted seams might have been of better than average quality. We cannot disregard this possibility entirely, but such investigations as have been made have failed to show any differences of such magnitude as might be expected to account for the wide difference in fracture records.

If we eliminate the effects of the material and accept the premise that construction practices were more or less uniform and subject to a corresponding degree of improvement as time went on, we can only account for the wide difference in the records of the two types of ships with the original details and the considerably greater effect of the changes in details and improved practices in the case of the all-welded ships by accepting the premise that the ships with riveted seams were far less susceptible to minor defects resulting from workmanship or construction practices and to square hatch corners, square cuts in sheerstrakes, continuously welded bilge keels, etc.

The foregoing discussion relates only to the comparative records for the all-welded vessels as against those for the ships with all of the shell seams riveted and it must be admitted that the records for these ships which had only one riveted joint at the gunwales or at the most only one at the gunwales and one in the upper deck on each side are nowhere near so con-

clusive. However, if there is any effect on performance of the introduction of a large number of riveted joints, then it is only reasonable to expect that there should be some effect due to only one or two such joints even if only to a considerably lesser degree. It is quite easy to establish just what is an all-riveted or an all-welded structure but it is impossible to define the different stages between these extremes as to when a structure might be said to progress from a riveted structure with some welding to a welded structure with some riveting. It is hard for me to believe that there is any magic point in this progression at which there is a complete flop-over from one set of conditions to the other. I believe that if we accept that the superiority of the fracture records of the ships with riveted shell seams is, to any degree, due to the riveted seams, then it is impossible to believe that the introduction of only one or two such joints can have anything but a beneficial effect, even if only to an insignificant degree.

I believe that Mr. MacCutcheon has done a very splendid job in his attempts to set forth all of the evidence pro and con on this very important subject and that even though his investigations have not produced any positive results, nevertheless they have given us all considerable food for thought. It is to be hoped, that as the many investigations progress and the service records accumulate, we will obtain a much clearer insight into this problem.

Discussion by Wendell P. Roop.

Wendell P. Roop is connected with Swarthmore College, Swarthmore, Pa.

The subject of this paper is more restricted than appears at first sight. Perhaps it can be paraphrased so: Does welding favor brittleness? The author concludes that we have no answer. My comment is in the vein: "If so, why?"

The inception of a progressing crack is a more elaborate process than was once thought. In laboratory specimens it is visibly apparent that a crack is present in a notched specimen soon after passing the nominal elastic limit; and does not the author have a standing offer to find cracks on any ship in service? It is an inadequate view of the matter to suppose simply that stress concentration causes cracks and that, once formed, they intensify the concentration and so lead to rupture. The choice does not lie between structures with cracks and those with none, but between brittle cracks and ductile cracks.

The question thus becomes this: Are the cracks in a riveted structure more likely to be ductile than in a welded structure? Since facts on this point are inconclusive, we must still depend on inference.

The usual thought has been that slip in riveted joints adds to the power of a structure to wash out localized effects, whereas in welded joints this power is wholly dependent on ductile flow in the metal. One might suppose that in the *Alexander* the riveted joints had suffered ankylosis, a common effect of old age.

The inclusion of Item 5 among the indisputable facts seems a trifle hasty. Will not slip generally occur first at those points where stress exceeds the capacity of the joint to resist it, and so shift the burden toward other points better fortified or previously less highly stressed?

The absence of laps or straps in flush seams of the sort usual in welding hinders contraction in width, but so long as the plate is free to contract

in thickness, triaxiality remains small.

There remains the possibility of a metallurgical effect on the brittle transition temperature. Grossman and MacGregor, last May, reported data to show that the metal in the welds is always more ductile than in the base plate. A small contrary effect was found at 1 in. from the center-line of the weld, the transition temperature being raised 5 to 20°. This seems a small price to pay for the benefits of welding, but its existence should not be ignored.

Discussion by David Arnott.

David Arnott is a retired Vice-President and chief Surveyor of the American Bureau of Shipping, now retained as a consultant. He is a Past-President of the American Welding Society.

The author makes a distinction between the inception of a crack and its propagation. It is axiomatic that every crack must have a beginning but a crack which merely starts but does not propagate is of no practical concern. There is no question in my mind but that welded ships are more susceptible to the inception of plate cracks than riveted ships at least during course of construction. When a plate cracked in the Yard in a riveted ship it was usually because of cold flanging, repeated furnacing during the fitting of shaped plates, abnormally high carbon, sulphur or phosphorous content or for some other satisfying reason. When we came to build welded ships, plate fractures kept occurring in the shipyards for reasons which were not so obvious. In the absence of any external loading the fractures could only be properly attributed to internal stresses incidental to the process of welding whether one prefers to call them shrinkage, thermal, residual or just plain locked-up stresses. Such failures were gradually reduced in numbers as the importance of adhering to proper welding sequences and technique became more fully recognized but that they could happen during course of construction was not a very good augury for the satisfactory behavior of the structure under service conditions. I have always felt that despite the apparent results of some of our welding research we were somewhat premature in discounting the importance of residual stress as a major contributing factor in our welded ship failures.

The structural failures of the riveted ships cited in the paper are not quite analogous to our welded ship failures. The *Oakey L. Alexander* managed to live up to her christian name for 31 years before breaking in two. I have not had an opportunity of examining her scantling plans but I would be very much surprised if the effective area of her strength deck would compare at all favorably with our modern standards for longitudinal strength. The *Leviathan* and *Majestic* were practically sister ships both of which experienced a partial failure of the upper structure after some years of transatlantic service primarily because of inadequate compensation in way of elevator and other openings in the strength deck in the vicinity of an abrupt discontinuity in the structure resulting from a line of expansion joints in the houses directly above. In short, one could point a finger at the reasons for the structural failures which have occurred on riveted ships. The failures on our completely welded ships, practically all new vessels, were much more mysterious. Experience has shown that once a crack started it was liable to spread with lightning rapidity across the deck and down the sides or across the bottom and up the sides until the structure had failed completely. Even where the crack originated in a member which was structurally unimportant such as thin plating

between hatches or a bulwark rail and which, although not included in the longitudinal strength calculation are subject to stress, the crack was liable to spread into the main hull structure if the light plating was welded thereto.

The comparative immunity of the Bethlehem-Fairfield ships with riveted shell seams has not been satisfactorily explained by any of the research projects which have so far been carried out. It is hoped that when the British experiments on actual ships, both welded and riveted, to which the author refers in his paper are finally completed additional light will be thrown on this important problem. The author seems to incline to the opinion that the only value of the longitudinal riveted straps or gunwale connections is to limit the propagation of cracks but I am not so sure that such is the case. Until such times as we have more information on the subject ordinary prudence will dictate the policy of installing crack arrestors in our new welded ships of the larger sizes as a precaution against complete failure. The *Oakey L. Alexander* and other riveted ships which broke in two were full of crack arrestors which obviously failed to function as such. The installation of these riveted crack stoppers in welded ships should be looked upon as a temporary expedient as it is no answer to the problem of how to build large *all-welded* structures which can be relied upon to be entirely satisfactory in service. Meantime, we are gaining more and more experience on welded ships in service and the policy of analyzing each and every structural failure hand in hand with a suitable program of fundamental research will undoubtedly go far toward providing a complete solution to our problems.

Mr. MacCutcheon has been unusually modest in submitting his conclusions. I agree with No. 2. So far as No. 1 is concerned I feel, for reasons already stated, that a welded ship is more susceptible to the inception of cracks than a riveted ship. With regard to conclusion No. 3, while I am prepared to admit that rivet slip could result in an increase of stress in critical portions of the structure it could just as easily result in stress reduction.

The author has been very fair in his handling of the extremely difficult subject of the relative incidence of cracking in ships' structures fabricated by riveting and welding, respectively. His lack of bias and the manner in which he has presented and discussed the various arguments pro and con are worthy of emulation by other writers of technical papers.

Discussion by John Vasta.

John Vasta, Naval Architect, Navy Department,
Bureau of Ships.

This paper has attempted to elucidate a very confusing subject. It appears however, that in going through the tortuous paths of direct and indirect evidences, hypothesis and postulates the author has not succeeded in adding the clarity desired. In fairness however, it should be noted that the pitfalls are many as the subject requires quantitative evaluation of too many elusive and somewhat obscure variables. This discussor finds it difficult to accept Conclusion 1 of the paper, and wishes to discuss two elements which have a bearing on the question of susceptibility to incipient cracking, namely, the matter of stress concentration and the effect of the residual stresses.

A welded structure, in my opinion, is more susceptible to incipient cracking than a riveted similar structure for the following two reasons:

(1) The process of welding *per se* delineates clearly the boundaries of a

geometrical discontinuity and is generally responsible for the formation of sharp notches. To illustrate, by welding all the component elements of a square hatch corner, for example, the acuity of this geometrical notch becomes well defined and established. This is not the case for a hatch corner structure assembled by riveting. It is apparent, therefore, that the acuity of a notch is more pronounced in a welded structure than in a similar riveted one. Since the magnitude of the stress concentration is a function of the acuity of the notch, the stress level at the base of such a notch should be higher in a weldment than in a comparable riveted assembly. We know that the inception of a crack depends largely on the stress level existing at the immediate region where the crack initiates.This means that the fracture stress is likely to be reached sooner in a welded structure than in a riveted structure of the same geometrical configuration.

(2) In addition to the severity of the stress concentrations existing at the base of a notch, the process of welding introduces a system of residual internal stress of high magnitude at the base of such notches. It is readily accepted that these residual stresses are non-existent in a riveted structure. However, we cannot admit as yet that the presence of residual stresses at the base of a geometrical discontinuity has no significant effect on the strength behavior of a welded structure. Recent published work* indicates that a tensile residual stress existing at the base of a notch has a significant effect on the fatigue strength of steel, whereas the same tensile residual stress present in unnotched plates plays no significant part. This is the first time that experiments point a guilty finger to the tensile residual stress as being important under certain conditions, fatigue and presence of notches. A weldment containing a geometrical discontinuity conceivably may have present tensile residual stresses of high magnitude in the three mutually perpendicular planes. This we know results in a triaxial stress system, a condition that, admittedly, theory tells us, should be avoided. The question to be asked here is whether or not a triaxial stress system could exist at a base of a geometrical discontinuity of a riveted structure. Obviously, the answer to this is in the negative. Hence, we should expect the welded structure to offer a higher probability to incipient cracking than a riveted assembly of similar details.

* The Welding Journal, 27 (8), Research Suppl., 421-s to 425-s (1948).

Author's Closure

The author of this paper is honored to receive discussions from those members of the shipbuilding profession who have contributed their comments. As stated in the paper, this study yielded very little in the form of positive findings. Some important factors are brought out in the discussion which add to the available information on the susceptibility to fracture of riveted and welded hull structures.

Captain Roop has presented several of his customary thought twisters based on the subject of the paper. He makes a very interesting statement which bears repeating: "the choice does not lie between structures with cracks and those with none, but between brittle cracks and ductile cracks." Mr. Arnott also supports this point with his comment that nonpropagating cracks are not important. Captain Roop has alluded to the author's statement that practically every welded ship contains numerous small cracks. Most surveyors of ship hull structure will support this observation; a fact which emphasizes the importance of nonpropagating cracks.

Captain Roop has questioned the in-

clusion of Item 5 among the indisputable facts. Perhaps the paper should have distinguished more clearly between the local and over-all effects of rivet slip. In Item 5 it is stated that "rivet slip under service loads may be accompanied by an increase of stress at critical locations in the structure." This statement referred to the calculations described in the paper and the calculations applied to over-all adjustments in the hull and not to local effects. It is mentioned elsewhere in the paper that local slippages may, and probably do, occur with corresponding readjustment of stress distribution. Detection of such redistribution in actual structures would yield important findings, but the selection of locations for study is attended by apparently insurmountable practical difficulties.

Mr. Brown has raised two questions with regard to the statistics in the paper which relate the presence of riveted crack arrestors to the incidence of cracks. The statistics include only Class 1 and 2 fractures and therefore correspond to Mr. Brown's data in this respect. The Bethlehem-Fairfield ships were included, however, and make up about ⅛ of the group without crack arrestors and a somewhat smaller proportion of those with arrestors. Mr. Brown kept the Bethlehem-Fairfield ships separate. To assist in rationalizing the differences Mr. Brown has graciously forwarded to the author a breakdown of the structural failure data included in his comment. On examining this breakdown one other difference was noted; Mr. Brown's tabulation lists casualties, whereas the paper under discussion lists fractures. Most casualties involved more than one fracture, the average being about three fractures per casualty.

The casualty data for the winter of 1944-45 were excerpted from Mr. Brown's tabulation. The incidence of casualties was found to be 70% higher for the ships with crack arrestors than for those without. This is good agreement with the data presented in the paper when it is realized that a wide dispersion is characteristic of statistics based upon such small samples and that the figures being compared are not coterminal.

It was reassuring to find that some additional pieces of evidence bearing on the subject of the paper had been included in the comments. Mr. Brown shows by his tabulation that improving the structural details on the all-welded ships resulted in a far greater improvement in structural performance than was brought about by a corresponding improvement in structural details for those ships which had riveted seams. (The items on Mr. Brown's tabulation have been numbered 1 to 5 for the purposes of this reply. Compare Group 2 with Group 1 and Group 5 with Group 4.) This type of comparison eliminates some of the variables which upset a direct comparison of Bethlehem-Fairfield's ships with all other Liberty ships. Questions of workmanship and steel quality are no longer important because the comparison is between the same ships with and without modification. There is, however,

Group	Type	Casualties per 1000 ship-months of service	
		All service up to July 1, 1948	Service from May 1, 1944, to July 1, 1948
1	All-welded with original details	16.2	3.7
2	All-welded with improved details	3.0	2.2
3	All-welded with improved details and crack arrestors	3.3	3.3
4	Riveted seams with original details	4.5	2.9
5	Riveted seams with improved details	2.3	2.0

one wild variable in this comparison, namely: the equality of operating conditions. During the winters of 1942-43 and 1943-44, the operating conditions for the Liberty ships were considerably more severe than during later winters. This was partly due to inclement weather conditions and partly due to the fact that many of the ships were operating to Murmansk, and in the North Pacific. The data were tested to see if all groups were equally affected by the severe weather. The tabulation resulted.

It will be noted that the tabular values, for operation up to July 1, 1948, do not agree exactly with those included in the comment. Mr. Brown has indicated corrections which have been applied to the revised figures above. It is clear that the service during the winters of 1942-43 and 1943-44 resulted in a much higher fracture incidence in some groups of Liberty ships. Since the number of various groups of vessels in operation was not constant throughout, this severe condition affected some groups more than others. Consider the last column for casualties from May 1, 1944, to July 1, 1948. Comparing the casualties per one thousand ship months of operation for Group 1 with Group 2, it can be seen that the fracture incidence was reduced to 60% by improving design details. Comparing Group 4 with Group 5, we find that the fracture incidence is reduced to 69%. The difference between 60 and 69% is no longer significant in light of the fact that one of the figures involved is based upon only four casualties. Here, again, we are confronted with the disappointing necessity of concluding that the evidence shows nothing significant regarding the relative incidence of cracking in welded and riveted ships.

Mr. Arnott has presented an important piece of evidence in pointing out that cracking during construction was much more widespread for the welded hulls than it had been for riveted hulls. This is a piece of direct evidence which was overlooked in preparing the paper. Mr. Arnott rightfully warns against predicting the incidence of failure of ships in service by making use of data from ships under construction. Everyone will agree, however, that the failure incidence during construction is a rough indicator of the direction of a trend and that a higher incidence of fracture in completely welded structures could be expected.

Mr. Vasta has included some very convincing explanations of why the welded structure should be more susceptible to cracking than the riveted structure. He has pointed out that welding creates worse notches and higher locked-in stresses. We are all attracted by the logic of such statements but similarly convincing arguments can be presented to support the opposing viewpoint. Although the logic is appealing we should be cautious in accepting these speculations even though equal consideration is given to the opinions supporting opposing views.

Considering the controversial nature of the subject, the discussors have taken an extremely objective viewpoint. This sort of healthy situation is bound to accelerate our understanding of the problem. The data presented in the paper reinforced by the data and reasoning included in the comments do not warrant a positive conclusion but provide an adequate basis for an opinion. The author holds the opinion that the incidence of cracking is higher in welded than in riveted structure. Even to this, a caution must be added: the opinion is based on existing structures; improvements in steel, in design, in workmanship or in weldability controls could bring the quality of welded structure up to or above that of the riveted structure.

ENGINEERING AND MARINE BOILERS.

ACKNOWLEDGMENT.

The following article by MR. G. W. KESSLER of the Babcock and Wilcox Co. is reprinted from the "Marine Journal" for March, 1949. Marine boilers are discussed in the light of overall plant considerations.

The design of a marine boiler is an engineering problem; as such, it involves the correlation of basic design data to meet the operating characteristics prescribed by the naval architect. The design solution follows the trend of any general engineering problem; that is, the present status is noted, the background is reviewed, and the approach to the immediate goal, or solution, is established.

In following this logical procedure in design solution it must be noted that marine boiler design is similar to both stationary and locomotive boiler designs—and differences are primarily a matter of application to specific requirements. Essentially these differences are due to considerable variations in the continuity of operation, available space, weight allotments, requirements for accessibility, type of fuel, firing methods and feedwater.

Any engineering problem can be likened to a broadbased pyramid with the base representing the starting level and the unsolved problems, and the apex, where there are no problems to be solved, the ultimate goal—perfection. However, there is no straight path to the goal, as it is necessary to stop often, consolidate the gains, review the background and chart the next advance. Consequently the working base, after each advance, is smaller than the preceding base and the path indicated may deviate considerably from that originally contemplated. In many developments the goal is never reached or even closely approached.

Goals in Marine Engineering.

In marine power plant design there are, in reality, three separate goals—all closely interrelated. The first goal is the improvement of the heat cycle and thus the thermal efficiency of the power plant; the second is improved economy for any heat cycle, that is, increased boiler efficiency and reduced fuel consumption; and the third is improvement in boiler design and decreased first cost.

The boiler designer, because of his developments in the use of materials, fabrication, design know-how, etc., cooperates in striving for the first goal although the steam pressures and temperatures selected are usually the result of an over-all economic survey. For any particular installation the selection of steam pressures and temperatures, well below the usual practice, may be justified economically, even though they restrict the maximum attainable thermal efficiency.

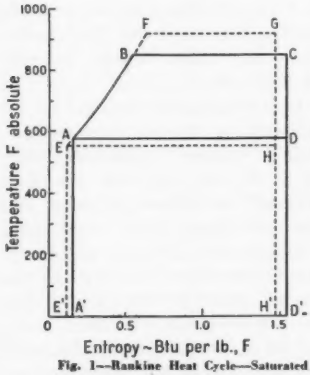
The boiler designer, however, is directly responsible for the attainment of the second and third goals. He must develop methods for improving boiler efficiency. This involves the application of better firing equipment and the most advantageous disposition of heating surfaces to reduce the exit gas temperatures. He also must improve design so as to assure continuity of operation, minimize maintenance, and reduce cost.

In design improvement, both functional and structural considerations must be reviewed and properly related. Often these are assumed to be inde-

pendent subjects, but this is not the case. Certainly, it is obvious that a superheater may have sufficient heating surface to deliver steam at a specified temperature (functional design) and yet the design may fail because of inadequate materials, high stresses and poor supports (structural design).

Heat Cycles.

Although the boiler designer cannot select the operating steam pressures and temperatures arbitrarily, it is important that he understand fully the various ideal heat cycles because of their influence upon boiler design. Typical Rankine cycles are shown in the temperature-entropy plots of Figs. 1 and 2.



LEGEND

	CONDITION — 200 psi STEAM 27" Hg VACUUM	CONDITION --- 450 psi STEAM 28.5" Hg VACUUM
Heat supplied–Area	A'ABCD'	E'EFGH'
Work done –Area	ABCD	EFGH
Heat rejected–Area	A'ADD'	E'EHH'
Eff. = Work done / Heat supplied	ABCD / A'ABCD'	EFGH / E'EFGH'

Fig. 1—Rankine Heat Cycle—Saturated Steam—Effects of Variation in Steam Pressure and Vacuum

These plots indicate the three main processes in power generation as well as the areas necessary to establish the cycle efficiency. The total area represents the heat supplied (by the boiler unit), the large bottom area indicates the heat rejected (to the condensers), and the smaller top area shows the heat absorbed (work done by the engine). The cycle efficiency, of course, is the ratio of the heat absorbed to that available.

The cycle efficiency can be improved by increasing the heat absorption while reducing the heat losses. It is apparent from Figs. 1 and 2 that the amount of work done and the cycle efficiency can both be increased by raising the high-temperature potential or lowering the low-temperature potential.

The high-temperature level can be raised by increasing the steam pressure or temperature; it usually is raised by increasing both, since the saturated steam temperature increases only from 388 to 705 F. when increasing the pressure from 200 to 3200 psi. Further, appreciable increases in pressure often are restricted by the requirements for thick and heavy pressure parts.

The use of superheated steam raises the high-temperature potential and thus increases the available work and the cycle efficiency. Higher initial steam temperatures also reduce engine exhaust moisture, thus improving the efficiency of the engine.

Advances in steam temperature are limited primarily by the available materials and quite often the oxidation temperature limits of the existing alloys are reached. When this situation occurs, the trend toward the use of reheated steam usually appears and for maximum cycle efficiency both the reheaters and superheaters are designed for the same exit steam temperature. If metallurgy then advances and steam

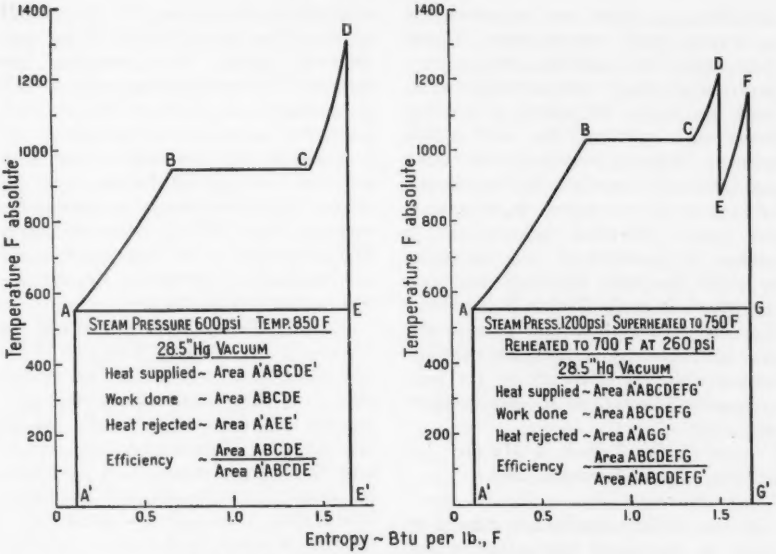


Fig. 2—Rankine Heat Cycle—Superheated and Reheated Steam

temperatures can be increased, the reheat installations generally become less attractive because of the additional piping and valving required. However, the recent sharp increases in fuel costs and the trend toward single-boiler, single-turbine stationary installations in an effort to eliminate excessive piping and valves have greatly encouraged the use of reheat.

The low-temperature potential is controlled by the temperature of the cold body which receives the rejected heat and thus the heat loss can be reduced by using a high vacuum in the condenser. Heat losses also can be reduced by using regenerative feed heating. This process utilizes bled steam from the engine to heat the feedwater and consequently less heat is rejected to the condenser.

Effect of Heat Cycle on Boiler Design.

The influence of the heat cycle upon boiler design is reflected by the trend toward higher steam pressure, steam temperature, and feedwater temperature. Further, although the desire for improved boiler economy necessitates the use of additional heat reclaiming apparatus—air heaters, economizers, or both—the use of high steam pressures also increases the demand. This is because the exit gas temperature from a

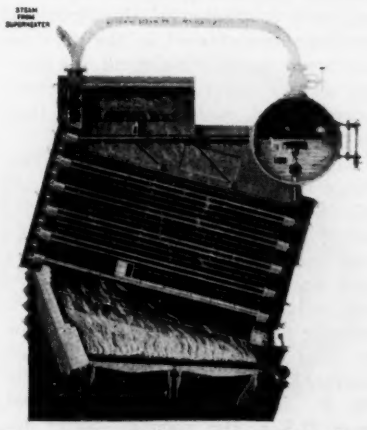


FIG. 3.—Hand-Fired Coal Three-Pass Header Type Marine Boiler with Superheater 200 psi, 50 F. Superheat.

boiler can, at best, only approach the saturated steam temperature. Therefore, since the saturated steam temperature increases with increased pressure, the boiler efficiency, at a given firing rate, will decrease with steam pressure. Thus, as heat cycle efficiencies are increased—particularly by the use of high steam pressures—it is imperative that additional heat-reclaiming surface be installed so as to maintain, or better, the boiler efficiency used with the previous cycle. If this is not done, there may be little or no gain in the over-all fuel consumption, since the increased thermal efficiency of the heat cycle may be offset by the lower boiler efficiency.

General Design Trends in Marine Boilers—Steam Pressures and Temperatures.

Marine boilers usually are classed as Navy or merchant types. The major differences in design are due to the wide variations in service requirements, evaporative ratings, size, and weight. The design changes made in both types during the past 30 to 35 years have resulted in functional designs which approach the ultimate goal—and each advance has been pyramided upon the satisfactory solution of the structural problems encountered in the previous functional design. As a result, the present marine boiler is smaller, lighter, more efficient, and more reliable than its predecessor.

The development of marine boiler design has followed closely the pyramid solution concept of the general engineering problem. Thirty-five years ago most marine boilers were of the three-pass, header-type, utilizing 4-in. tubes and delivering saturated steam at a pressure of approximately 200 psi. For the heat cycles then in use—200 psi saturated steam with approximately 200 F. feedwater temperature, and about 27.5-in. Hg. vacuum—boiler designs were improved to increase the boiler efficiency. The first improvement resulted in the substitution of a cluster of four 2-in. tubes for each 4-in. tube formerly used, thus doubling the amount of heat-absorbing surface in a given space. Air heaters and economizers also were used on occasion.

A change was then made in the heat cycle and, as a result, boilers were developed to deliver steam at a temperature of about 450 F. (approximately 50 F. superheat) the limit which could be handled satisfactorily by the reciprocating engines in use.

In the 20's an important event in marine boiler design occurred. This was the general acceptance by the industry of the small-tube single-pass header and drum-type boilers, which had, as their forerunners, the small-tube drum-type boilers used primarily in naval services during the first World War. Consequently, small, lightweight, compact boiler units, having good efficiencies and low resistance to gas flow, were available and the way

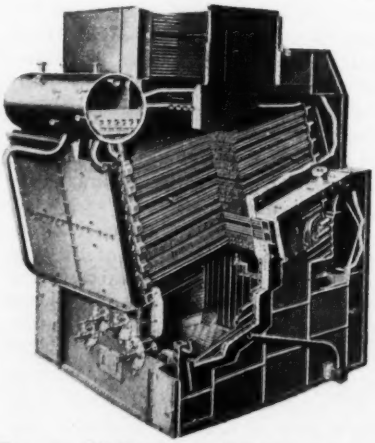


FIG. 4.—Oil-Fired Single-Pass Header-Type Marine Boiler with Superheater and Air Heater—450 psi, 750 F.

was paved for the subsequent improvement of, and changes in, the heat cycles.

The first of these improvements in the superheat heat cycle, occurring in

the early 30's, was of considerable magnitude, resulting in steam pressures and temperatures of 450 psi and 700-750 F. for merchant boilers and 400 psi, 650-700 F. (closely followed by 600 psi and 700 F.) for naval boilers.

In the middle 30's the efficiency of the heat cycle was improved further when the Navy decided to use steam temperatures of 850 F. At the same time a number of merchant boilers were built for steam pressures and temperatures of 600 psi, 825 F.; 625 psi, 910 F.; and 835 psi, 910 F.

Pilot reheat installations also were developed for naval and merchant services and were designed for comparable steam temperatures at the superheater and reheater outlets. However, in the naval installation the superheated and reheated steam temperatures were limited only by the available commercial alloys, while in the merchant installations lower temperatures were used to facilitate the use of carbon steel tubes.

Improvements in Marine Boiler Design.

The improvements and changes in the heat cycles greatly influenced boiler design and led to improved economy and the betterment of both functional and structural designs. As a result, water-cooled furnace walls were developed, which facilitated high furnace heat release and high average absorption rates and precluded excessive localized heat input to the furnace rows of boiler tubes and excessive furnace brickwork maintenance. High-capacity firing equipment also was designed to effect reductions in boiler size and weight—and the combustion efficiency of the firing equipment was improved to minimize heat losses.

Heat-reclaiming equipment became mandatory in order to obtain the desired high boiler efficiencies—economizers for naval installations (where low-pressure exhaust-steam is used for feed heating and maneuvering is rapid) and either air heaters or economizers on merchant jobs, depending upon the feed heating system. The majority of merchant installations were fitted with air heaters, because regenerative feed heating, resulting in feedwater temperatures of 300-350 F., was prevalent. Since the exit gas temperature leaving an economizer can never be less than the entering water temperature, the use of high feedwater temperatures restricts the boiler unit efficiency when using an economizer. This is not the case with an air heater where the uptake gas temperature can, except for corrosion considerations, approach the inlet air temperature. The use of regenerative feed heating is desirable, of course, because it decreases the cycle's heat loss.

Air-cased units were developed to allow operation with high gas pressures throughout the unit, prevent gas leakage into the fireroom, minimize heat losses, and, in the case of naval vessels, protect the personnel from enemy gas attack.

The high-rating, small, light-weight boiler units also necessitated considerable improvements in the steam drum baffles. Advances were made which improved not only steam separation but also the circulatory system.

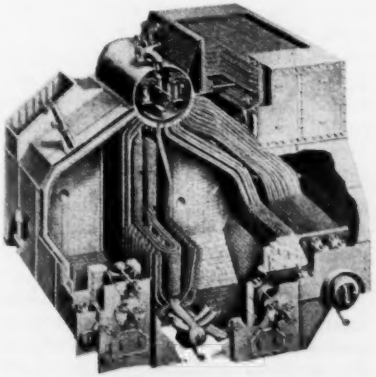


FIG. 5.—Oil-Fired Two-Furnace Single-Uptake Controlled Superheat Type Marine Boiler with Economizer—600 psi, 850 F.

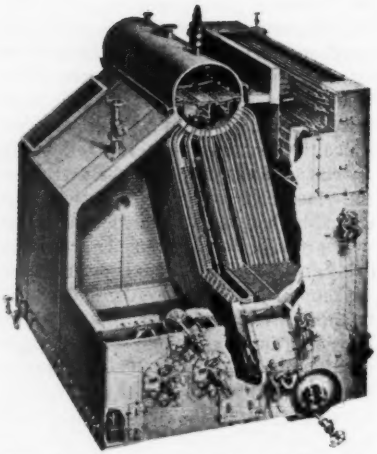


FIG. 6.—Oil-Fired Two-Drum Type Marine Boiler with Superheater and Economizer—600 psi, 825 F.

The use of high steam temperature occasioned a demand for temperature control and this was accomplished by means of two-furnace boiler units (Ref. Fig. 5) or steam drum attemperators.

These important design developments —water-cooled furnace walls, high-capacity firing equipment, heat-reclaiming apparatus, air casings, steam separation, and temperature control—were supplemented by a number of other design improvements, such as light-weight alloy casings, water-cooled superheater supports, water-cooled gas baffles, furnace refractory arrangements, etc., all of which resulted in tremendous strides toward the goal.

Marine Boilers for Great Lakes Vessels—Old, Reboilering and New Construction.

The trend of boiler design for Great Lakes steamers has followed closely the over-all trend of marine boilers and the pyramid solution concept of engineering. Thirty-five years ago most Lake vessels were driven by multiple-expansion reciprocating engines using saturated, or slightly superheated, steam at 200 psi pressure. The boilers were of the Scotch or the three-pass header-types and were coal fired by hand. Today, marine boilers for Great Lakes bulk cargo carriers can be classed in two general categories—boilers for replacements in old ships, and boilers for new construction.

Boilers for replacements in old vessels usually are designed for the original operating conditions—200 psi steam pressure, 450 F. steam temperature, and 220 F. feedwater temperature. However, on occasion, engine changes are made to facilitate the use of more than 50 F. of superheat. Although these new boilers operate in the original heat

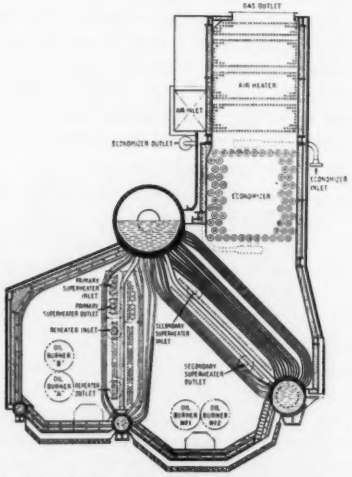


Fig. 7—Oil-Fired Two-Furnace Single-Uptake Type Marine Boiler with Superheater, Reheater, Economizer and Air Heater—1235 psi, 750 F. and 220 psi, 750 F.

cycle, they have embodied in their design the developments discussed in connection with the over-all marine boiler trend. Thus the original Scotch and 4-in. tube, three-pass, heater-type boilers are replaced by smaller, lighter-weight, more efficient single-pass boilers—usually of the header-type—fitted with superheaters, air heaters, and spreader stokers.

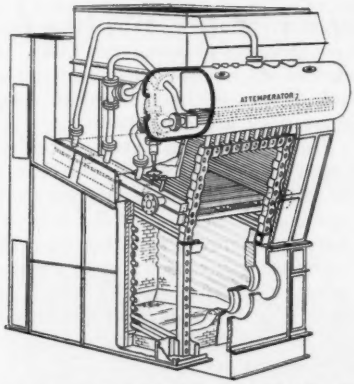


Fig. 8—Oil-Fired Single-Pass Header-Type Marine Boiler with Attemperator for Steam Temperature Control—625 psi, 910 F.

The substitution of spreader stokers for hand-fired grates led to considerable reduction in boiler size, since both the firing rate per sq. ft. of grate area and the length of the grate were increased. The use of water-cooled furnace walls also helped to decrease boiler size considerably because the amount of "cold" or radiant heat-absorbing surface in the furnace rows of tubes could be reduced. The installation of air heaters provided heated air, thus improving the combustion efficiency. Further, the additional heat-absorbing surface effected a reduction of about 175 F. in the gas temperature leaving the boiler tube bank and thereby increased the boiler efficiency about 5 per cent.

For new construction the boiler designs are almost identical with those used for reboilering, with the general exception of changes in pressure part thicknesses and the size of the superheater to facilitate the delivery of steam at 450 psi pressure and 750 F. In addition to the increased steam pressure and temperature generally used in new construction, the power for propulsion has increased from approximately 2000 I.H.P. to about 4000

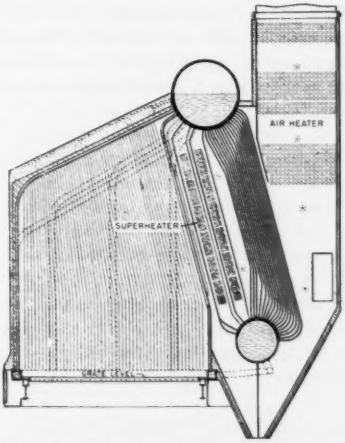


Fig. 10—Stoker-Fired Two-Drum Type Marine Boiler with Superheater and Air Heater—450 psi, 750 F.

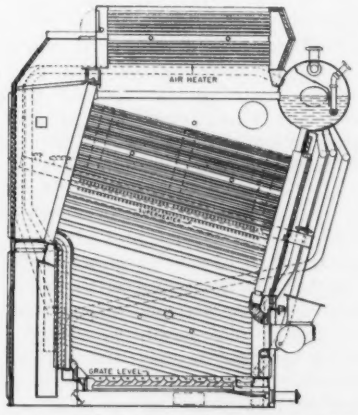


Fig. 9—Stoker-Fired Single-Pass Header-Type Marine Boiler with Superheater and Air Heater—200 psi, 450 F.

S.H.P. and the geared-turbines exhaust into surface condensers. Thus very little evaporated make-up water (about 2 per cent) is required as compared to the amount (approximately 85-90 per cent) necessary when using jet condensers with the reciprocating engines. Consequently, economy is improved, since considerably less feedwater treatment is required, and boiler blow-down and, therefore, heat loss is reduced.

For the prevailing power requirement (about 4,000 S.H.P.), surveys by the operators indicate that the use of steam pressures and temperatures of approximately 450 psi and 750 F. result in the greatest over-all economy. If, however, future economic conditions dictate the use of greater power, serious consideration must be given to higher steam pressures and temperatures.

Effect of Space on Boiler Design.

In possible future installations certain influences—furnace designs for coal firing and space available for boiler installation—may tend to limit the maximum steam temperature. Furnaces must be designed to keep the furnace gas temperature below the ash initial deformation temperature in order to prevent slagging. This, consequently, results in relatively low gas temperatures entering the superheater and necessitates a considerable amount of superheater heating surface, if high steam temperatures are required. However, because of space conditions, the amount of surface which can be installed may be limited unless design innovations are developed.

Space conditions also affect the length of gas travel from the grate to the tube bank in both header and drum-type boilers. Because of the relatively short flame travel, resulting from size restrictions imposed upon marine boilers, the firing rates per sq. ft. of grate area are limited to about 50-60 per cent of those common to stationary installations.

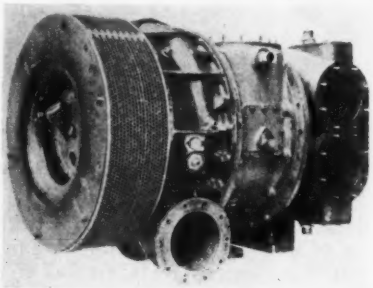
Conclusion.

The progress of marine boiler development for Great Lakes vessels is typical of the general over-all marine trend and important strides have been taken toward the ultimate goal—improved heat cycle, minimum fuel consumption, and better boiler design.

A BRITISH WATER-COOLED TURBO-BLOWER.

ACKNOWLEDGMENT.

This article describes a new exhaust-gas Turbo-blower for pressure charging oil engines. It is reprinted from "The Marine Engineer and Naval Architect" for February, 1949.



Type TS200 Napier turbo-charger fitted with an experimental filter.

Since 1945 development has been proceeding at the Acton, W.3, works of D. Napier & Son Ltd., famed aircraft engine builders, on a range of exhaust-gas turbo-blowers for the pressure-charging of medium-power marine, stationary and railway traction oil engines. Nearly a year ago two prototype sets were fitted to a Vee-type 12-cylinder engine of English Electric make and this has now completed a good deal of running at the Rugby works of the parent firm. Further testing and development has been carried out at Acton but series manufacture will be made at Liverpool.

Universal applications.

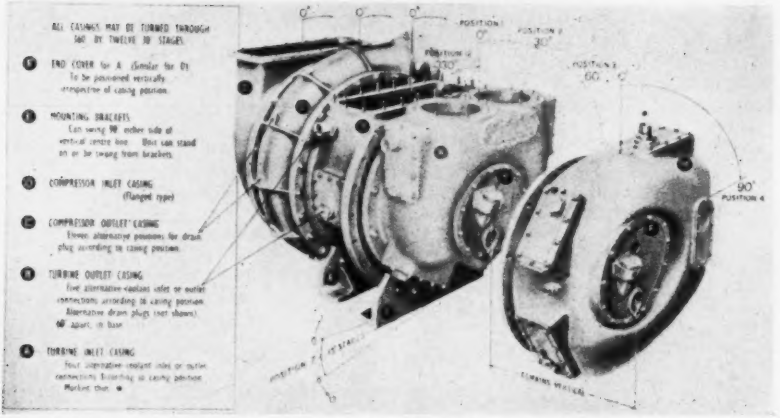
Each size of blower, as detailed in Table I, is being offered with a range of alternative turbine wheels and associated shroud assemblies, three alternative types of turbine inlet casing, and two different types of compressor inlet casing. Because of this, the production process is not a matter of producing four sets of main components differing only in size, but one of producing some 15 different major components for each separate unit, making some 60 different components in all. Thus, the production process resolves itself into one of batch-producing the various components to meet the particular build requirements demanded by different users; these build requirements vary continually according to the pattern of current sales. Production of the two smaller sizes is now proceeding at the Liverpool works, and is in the process of being extended to include the larger sizes.

The turbine and compressor castings form the main material mass of the assembled blower units; and they, together with turbine blade production, constitute the main bulk of the production task, at least in terms of handling and factory space. All the turbine inlet and outlet casings are sand castings of Meehanite, and are received from the foundry with all cores, volutes and coolant and air-bleed passages fully formed.

From this beginning, the completion of the production task is a straightforward machine-shop batch process: the various faces, registers and flanges for the gas entry and outlet ports, the

TABLE I.

Unit Type No.	Free Air Consumption at 5lb. per sq. in. Boost (cu. ft. per hour)	Engine Power Output per blower unit—B.H.P.	
		Without Turbo Blower	With Turbo Blower
T.S. 100	52,000— 85,000	180– 300	270– 450
T.S. 200	85,000—135,000	300– 500	450– 750
T.S. 300	135,000—215,000	500– 800	750–1,200
T.S. 400	215,000—340,000	800–1,400	1,200–2,100



Showing the universality of mounting and installation of the Napier water-cooled exhaust-gas pressure-charging turbo-blowers.

water jacket inspection plates, the oil reservoir and cover plates, the locating faces for the oil pump and bearing assemblies and the circular flanges by which the casings are assembled coaxially together, are all machined by normal turning, boring, milling and grinding practice. The drilling of the various holes for the studs, bolts and spigots completes the task of preparing the casings for the assembly shop. The shrouds that surround the turbine wheel are made of heat-resisting steel and are cast in threes, already flanged. Thus, the processing of these components in the machine shop consists of roughing, parting-off and finishing. The outer and inner rings of the stator nozzle assembly are similarly formed.

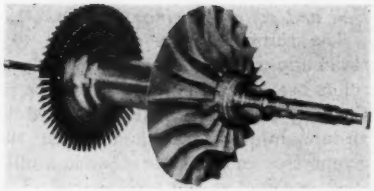
The filter and silencer is a fabricated item, assembled and filled with either of its chosen filter elements in the detail shops. The flanged type of compressor inlet, together with the compressor outlet casings, is a sandcast aluminium alloy casting with all cores, volutes and air passages fully formed in the foundry stages, as for the turbine casings described. The machine work also follows closely upon that required for the turbine parts and consists mainly of flange-facing operations and the drilling of stud and bolt holes for the axial assembly flanges. One operation of a more specialized character is carried out on the compressor outlet casing. This consists of forming the internal radius at the center, the work being done by a special copying head attachment fitted to a vertical boring machine.

Rotor assembly.

The turbine shafts are machined from bar steel and are ground over all diameters and shoulders. The splines for locating and driving the impeller, the slots for driving the turbine disc and the threads for the retaining nuts at the ends are machined on the solid shafts by milling and hobbing machines.

Impellers are received roughly formed in aluminium-alloy forgings, the blades being milled, formed, heat treated and subjected to pressure bending in the machine shops.

The turbine discs are made from austenitic heat-resistant steel, and the turbine wheels are assembled with separate turbine blades attached to the periphery of the disc by fir-tree root fixings. The discs begin life as forged "cheeses" and are bored, turned and



Rotor assembly showing turbine disc and blades, the blower impeller, and the common shaft. This impeller is 9¼ in. dia.

finished by normal lathe and grinding practice. The separate blades are individually machined from austenitic bar material, being milled and form milled on a special four-at-a-time hydraulically-clamped jig attachment on the bed of the milling machine.

All blades are finished by a series of grinding processes and are finally polished to a mirror-like surface; and all impeller discs and turbine wheels are individually dynamically balanced on an Olsen balancing machine. This machine records the amount of circumferential location of any out-of-balance weights on the disc or wheel while it is revolving at high speed on the testing spindle.

Ratings and construction.

These Napier turbo blowers were designed in general to pressure-charge at about 5 lb. per sq. in. above atmospheric, but, subject to engine design, can be run continuously with a delivery pressure of 7¼ lb., and up to 8½ lb. for periods up to 60 min. Their general characteristics are given in Tables I and II.

In principle, these Napier pressure-charges do not differ from those of other makes which have been applied. The set comprises a single-stage axial-flow exhaust gas turbine driving a centrifugal air compressor. It may be considered first as five separate portions: turbine inlet casing, turbine outlet casing, compressor inlet casing, compressor outlet casing, and the rotor assembly. All these parts can be picked out in accompanying illustrations; and they are bolted together to house the common shaft for the turbine wheel and air impeller.

The turbine inlet casing is a substantial casting of heat-resisting Meehanite and provides an exhaust-gas inlet to the turbine with a two, a three or a four entry flange that communicates with a ring of nozzle vanes. A bearing support is formed in the casing and an extension of the casing forms an oil reservoir. The turbine outlet casing, which forms the support for the whole unit, is also of heat-resisting Meehanite. The turbine inlet and compressor outlet casings are bolted, one on each face, to the turbine outlet casing. The two robust support brackets are also bolted to this casing to provide a balanced mounting point for the whole unit.

A sea-water-resisting aluminium alloy compressor outlet casing is used for the casting which contains the air inlet passages and delivery volute. It carries

TABLE II.

TYPE T.S. 100

Net Dry Weight—390lb.	Normal (50% S/C)	Maximum (continuous)	Maximum (1 hr. Limit)
Turbo blower R.P.M.	17,000	20,000	21,500
Boost Pressure	5	7¼	8½
Blower Pressure Ratio	1·35 to 1	1·5 to 1	1·6 to 1
Turbine Entry Temp.	500°C	600°C	650°C
For Naturally-aspirated Rating between 180 and 300 B.H.P.			

TYPE T.S. 200

Net Dry Weight—650lb.	Normal (50% S/C)	Maximum (continuous)	Maximum (1 hr. Limit)
Turbo blower R.P.M.	13,500	15,750	17,000
Boost Pressure	5	7¼	8½
Blower Pressure Ratio	1·35 to 1	1·5 to 1	1·6 to 1
Turbine Entry Temp.	500°C	600°C	650°C
For Naturally-aspirated Ratings between 300 and 500 B.H.P.			

TYPE T.S. 300

Net Dry Weight—1,100lb.	Normal (50% S/C)	Maximum (continuous)	Maximum (1 hr. Limit)
Turbo blower R.P.M.	11,000	13,000	14,000
Boost Pressure	5	7¼	8½
Blower Pressure Ratio	1·35 to 1	1·5 to 1	1·6 to 1
Turbine Entry Temp.	500°C	600°C	650°C
For Naturally-aspirated Ratings between 500 and 800 B.H.P.			

TYPE T.S. 400

Net Dry Weight—1,850lb.	Normal (50% S/C)	Maximum (continuous)	Maximum (1 hr. Limit)
Turbo blower R.P.M.	9,000	10,500	11,500
Boost Pressure	5	7¼	8½
Blower Pressure Ratio	1·35 to 1	1·5 to 1	1·6 to 1
Turbine Entry Temp.	500°C	600°C	650°C
For Naturally-aspirated Ratings between 800 and 1,400 B.H.P.			

one of the rotor bearings and an extension forms an oil reservoir. A vaned diffuser is assembled between this casing and the turbine outlet casing. There are alternative types of inlet casing; an intake incorporating an air filter and silencer or an intake with a large rectangular flange to communicate with ducting. The air filter can have alternative filtering elements, such as the Vokes type or the oil-wetted ferrule pattern.

The rotor assembly consists of a single solid steel shaft with a bladed turbine wheel mounted at one end and a centrifugal air impeller at the other. The heat-resisting materials used for the construction of the turbine wheel and all turbine parts are claimed to ensure long life at the high operating temperatures encountered. The air impeller, an aluminium-alloy forging, is of the single entry type with 17 radial vanes, and has a labyrinth-type air seal machined on its rear face. This seal is so positioned that the axial thrust balances out to a very small value.

Exhaust gases are directed on to the blades of the turbine wheel by a number of nozzle vanes. The blades of the turbine wheel are fashioned in such a manner as to provide the correct gas incidence angles at all sections of the blades, thus ensuring low losses and high efficiency. After passing the turbine wheel the gases leave the unit through the large rectangular aperture in the turbine outlet casing.

Rise in temperature in the turbine casings due to the passage of the exhaust gases is prevented from passing to the compressor assembly or the turbine bearing by water cooling both turbine casings. An internal tapping from the delivery volute in the compressor outlet casing provides air for cooling the front face of the turbine disc, while a designed leakage from the impeller labyrinth seal cools the rotor shaft and the rear face of the turbine disc.

Ball and roller bearings used.

The rotor shaft is carried in two resiliently-mounted high speed ball and roller bearings, which are a feature of the Napier design. One is located in an aluminium-bronze housing at the compressor end and the other in a mild steel housing at the turbine end. The former locates the shaft and takes the small axial thrust, while the latter can move longitudinally to permit the axial growth of the unit due to thermal expansion. Correct alignment of the bearings is ensured by the provision of spigots between the individual casings.

The bearings are the only parts which require lubrication. Each bearing is lubricated separately by its own oil pump, which is mounted on the shaft outboard of the appropriate bearing. Oil from the bearing is prevented from leaking into the delivery air stream and exhaust gases by air pressurized oil seals, the air for this duty being tapped off from the delivery volute and led to the seals by internal passages in the casings. The lubrication system is designed to comply with Lloyd's specification in that it will operate satisfactorily with the turbo blower permanently tilted at 15 deg. in any direction, and also at a temporary tilt of 22½ deg., such as might be experienced in marine installations during a roll.

In order to avoid the possibility of the exhaust pulse of one cylinder affecting the scavenging of the cylinder that has previously fired, alternative turbine inlet casings, with twin, triple, or quadruple gas entries are available —in which the ducts from the cylinders are kept apart right up to the turbine wheel, each duct feeding a separate group of nozzles. All gas inlet connections are circular and (with the exception of the type T.S. 100 blowers) they are counterbored to take a piston ring type of expansion joint. Studs, nuts and lock washers in heat-resisting material are supplied for the whole of the gas inlet and outlet flanges.

RESISTANCE WELDING OF JET ENGINES.

ACKNOWLEDGMENT.

In this article by MR. H. E. LARDGE, reprinted from The Welding Journal for March, 1949, the application of spot, stitch and seam welding to jet engines is described. Early experiences are reviewed leading up to present-day applications and techniques.

Introduction.

One of the foremost engineering developments of the present time is that of the gas turbine engine for aircraft propulsion and it is perhaps not generally realized how important a part resistance welding plays in the manufacture of such engines. One of the advantages of the turbine engine, as originally conceived, was that it was lighter in weight than a normal reciprocating engine of equivalent output. This result was obtained, in part, by the use of sheet metal for the construction of certain main sections of the engine (such as the combustion and exhaust systems) and the resistance welding processes of spot, stitch and seam welding are extensively used in this new field of engineering construction.

In order to appreciate the conditions under which parts and assemblies fabricated by such means must operate, it is necessary briefly to explain the working of the jet engine, which is comprised of four main items, namely:

1. The centrifugal compressor
2. The combustion system
3. The turbine
4. The exhaust system

Air from the high-speed compressor is led to the combustion chambers where a continuous supply of atomized liquid fuel is burnt. The resultant gases of combustion, now at high velocity, pass to the turbine (which drives the compressor) and then through the exhaust unit and jet pipe to atmosphere emerging in the form of a high velocity jet. From this it will be understood that the operating temperatures of the parts comprising the combustion chambers and the exhaust system (with which we are concerned) are high and entail the use of special heat-resistant alloys. In addition, high dynamic stresses are imparted to these components by high-frequency pulsations of considerable magnitude present in the air stream. It will, therefore, be appreciated that the application of the resistance welding processes to the manufacture of such assemblies was not accomplished without some difficulty.

On the original Whittle design resistance welding was not used to any great extent, but when the production of jet engines in quantity was envisaged it was obvious that resistance welding, and seam welding in particular, had many advantages to offer in the way of rapid and efficient production when once the difficulties inseparable from such an application had been overcome. This, at any rate, was the opinion of the engineers of the firm of Joseph Lucas Ltd., Birmingham, who were invited to collaborate with engine firms concerned with the development and manufacture of jet engines for the British Government, and subsequent experience has proved this view to be correct.

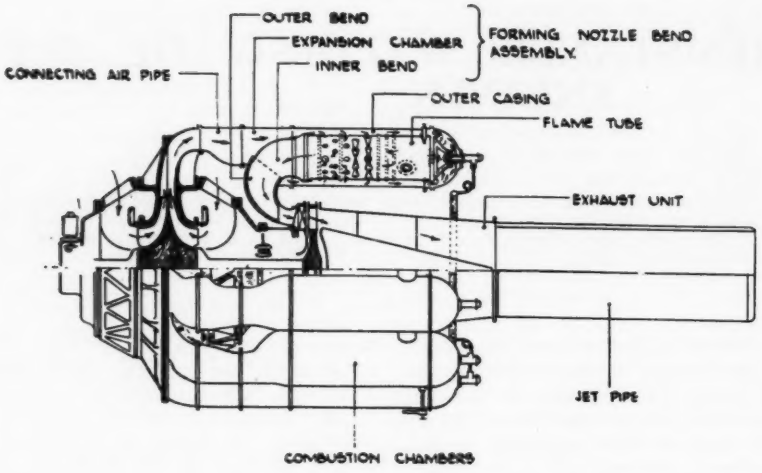


Fig. 1 Diagrammatic view of W.2.B. "reverse flow" engine

Early Developments and Experiences.

There are two main types of jet engines, namely the "reverse flow" and the "direct flow" types. Early engines were of the "reverse flow" pattern and Fig. 1 is a diagrammatic view of the Fower Jets W.2.B. design, later to be developed into the Rolls Royce "Welland." On this engine the combustion chambers, 10 in number, are cylindrical in form and are grouped around and parallel to the main axis of the engine. Each chamber consists of an outer, or air casing, and an inner member, or flame tube, inside which combustion takes place. The air casings operate at comparatively low temperatures and are joined together by interconnectors for pressure equalization to form one assembly (Fig. 2).

The casings were originally made from 18-8 stainless steel sheet (0.0164 in. thick) for purposes of resistance to corrosion. Each casing was made by "wrapping" and then longitudinally stitch welding along a narrow overlap, leaving the welded region similar in thickness to the metal. On engine tests, catastrophic fatigue failures were experienced along the longitudinal welds. Seam welding on a wider overlap was introduced, with some improvement in results, but the failures still persisted, sometimes after only a few minutes running.

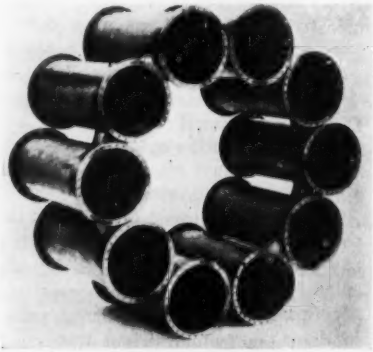


FIG. 2.—Air casing assembly.

FIG. 3.—Fatigue failure at seam weld.

An oscillographic check revealed high-frequency pulsations in the air stream which were inducing high dynamic stresses at the notch along the weld, thus allowing the fatigue strength of the metal to be exceeded locally and resulting in such typical failures as shown in Fig. 3. A change of material to mild steel overcame this difficulty and is accounted for by the higher internal damping capacity of this material allowing the energy loss to be more generally absorbed in the casing. Protection from corrosion is given by aluminium sprayed coatings.

The flame tubes (Fig. 4) are made from a specially developed sheet material called Nimonic 75, basically an 80-20 nickel chromium heat resistant alloy with high mechanical and fatigue strengths at elevated temperatures. The

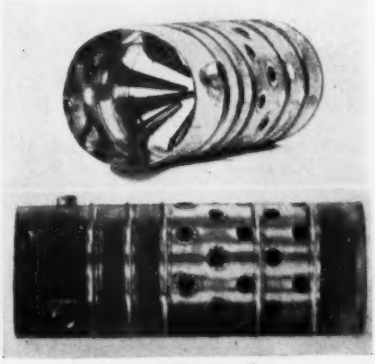


FIG. 4.—Flame tubes.

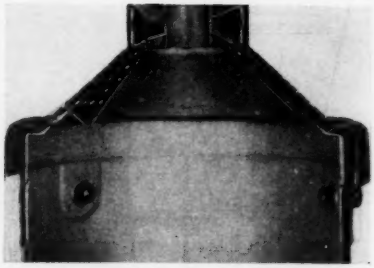


FIG. 5.—Flame tube head.

flame tube is made by wrapping and longitudinal seam welding, the strengthening bands being circumferentially welded to the flame tube. The stub pipes 16 per flame tube, are spot welded in position and the flame tube head (Fig. 5), which is of pressings of stainless steel and Nimonic 75, is also constructed by spot welding.

The discharge nozzle assembly, which forms one of the major units of the engine, provides a good example of the marrying of resistance and fusion welding (Fig. 6). This assembly, part of whose purpose it is to lead the combustion gases to the turbine, is made from resistance-welded stainless steel pressings and solid flanges machined from centrifugally spun austenitic steel cylinders. The expansion chambers, 10 in number, are made by stitch and seam welding half pressings together and these in turn are fusion welded to outer bends. The whole assembly is built up in a welding fixture or manipulator and fusion welded together.

FIG. 6.—Discharge nozzle assembly.

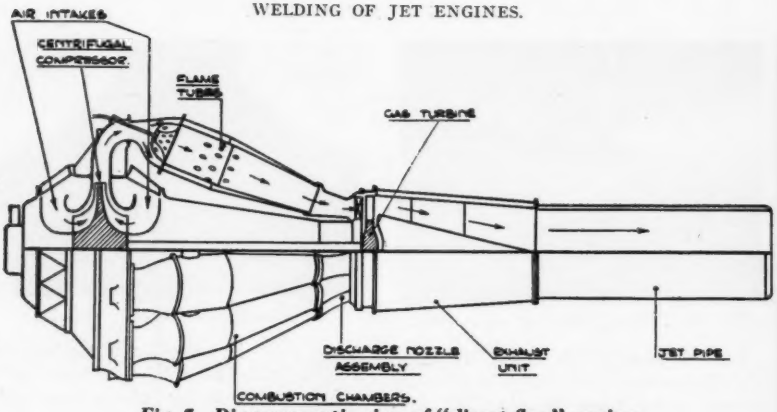


Fig. 7 Diagrammatic view of "direct flow" engine

The inner bends, which actually lead the gases to the turbine entry, are made from parts of thin Inconel sheet, stitch and seam welded together.

Later Engines.

As the exhaust units and jet pipes of either the "reverse flow" or "direct flow" types of engines are very similar in construction, it is proposed at this stage to say something about the manufacture of combustion chambers for some of the "direct flow" engines and a diagrammatic view of this kind of design, as exemplified by the Rolls Royce Derwent I engine, is shown in Fig. 7.

The air casing (Fig. 8) is of combined cylindrical and conical form and is again constructed from mild steel sheet by wrapping and seam welding. The stainless steel interconnector parts are stitch welded to the body prior to it being sprayed with aluminum. The sealing ring at the small end of the casing is of heat-resistant alloy and is seam welded to the mild steel body. The expansion chamber, which is shown bolted to the casing, is made from two odd-shaped half pressings of mild steel stitch welded together, the pressed end flanges being attached by the same process. The flame tube assembly (Fig. 9) is fabricated throughout by resistance welding and with the exception of the outer pressing of the flame tube head, the whole assembly is made from Nimonic 75.

The flame tube for the very successful Rolls Royce Derwent V engine, also of the "direct flow" type, is made throughout of Nimonic 75, as several fatigue failures had been experienced on the stainless steel part of the Derwent I flame tube head. This flame

FIG. 11.—Air casing

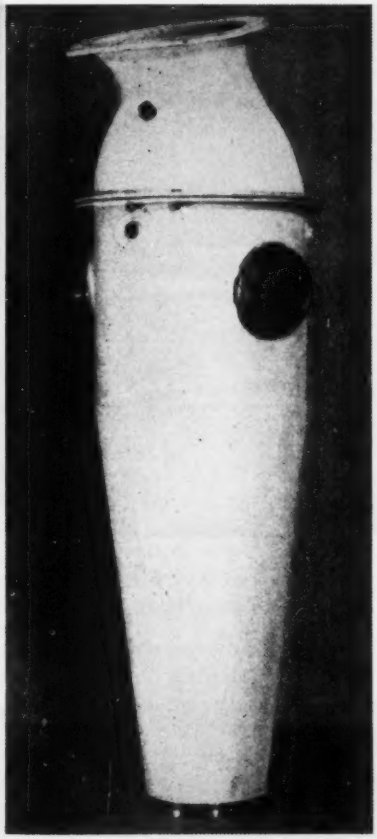


FIG. 8.—Air casing and expansion chamber.

tube (Fig. 10) provides an excellent example of the application of resistance welding to a dynamically loaded structure of unusual design and material. The construction of the air casing (Fig. 11) follows similar lines to those already dealt with and it is not proposed to go into detail about this.

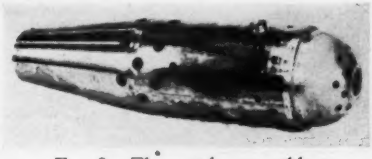


FIG. 9.—Flame tube assembly.

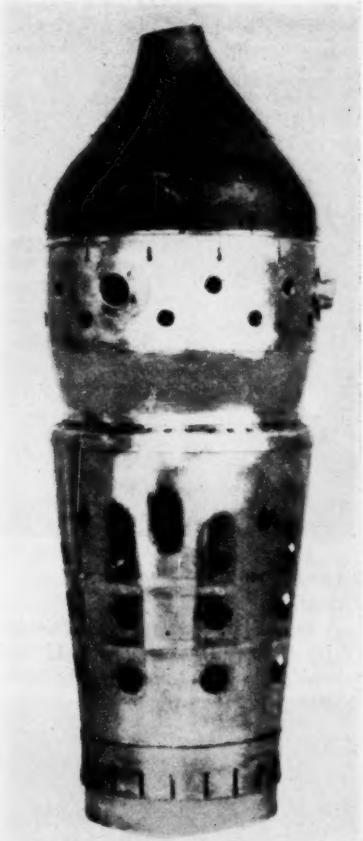


FIG. 10.—Derwent V flame tube assembly.

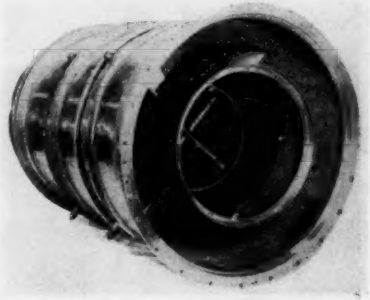


FIG. 12.—Exhaust unit, part-finished assembly.

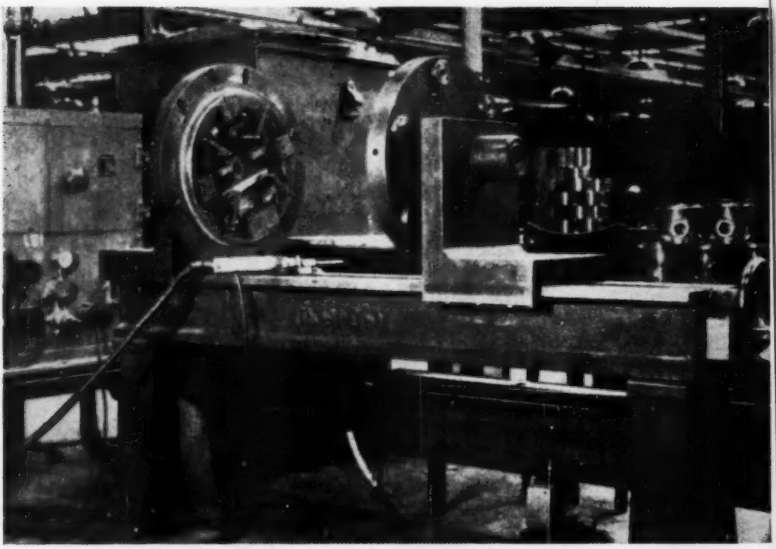


FIG. 13.—Tack spot-welding fixture for outer cone.

Exhaust Unit and Jet Pipe Manufacture

A typical exhaust unit for this type of jet engine is shown in Fig. 12. This assembly is located immediately behind the turbine and is very interesting from a resistance welding viewpoint. The truncated outer cone of stainless steel is longitudinally seam welded and is tack spot welded to the solid flanges

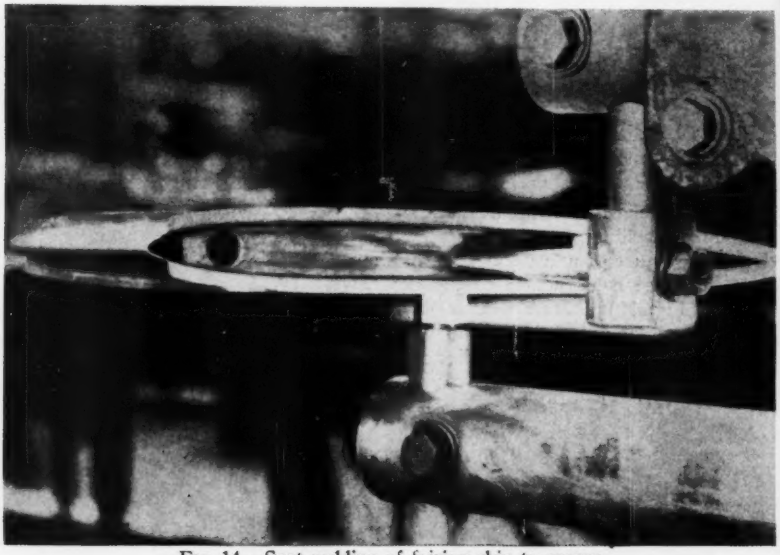


FIG. 14.—Spot welding of fairing skin to spacer.

FIG. 15.—Jet pipe assembly.

at each end in a fixture with internal expanding electrodes (Fig. 13) prior to the flanges being seam welded. The spacing bands are then seam welded to the outside of the cone, provision having to be made for distortion by leaving some machining allowance on the flanges.

The inner cone, whose purpose it is to streamline the gas flow behind the turbine disk, is also made from stainless sheet seam welded as far down as practicable, the remainder of the overlap being stitch welded and a machined tip is then stitch welded into the end to complete the conical form. A stiffening cone for carrying the cross-support bearings is spot welded inside the inner cone and a machined flange is stitch welded to the open end.

The cross tubes, which support the inner cone, have to be shielded from the gas stream to avoid losses due to turbulence and this is done by four fairings which cover the tubes in the annular space between the inner and outer cone. These fairings are made by spot welding an outer skin of Nimonic 75 to four stainless steel spacers (Fig. 14).

The last assembly to be described is the jet pipe (Fig. 15). This is made by joining a number of seamwelded sections of stainless steel to form a cylinder of the desired length. Spacing bands, similar to those on the exhaust unit outer cone, are seam welded to this cylinder which is then covered with heat insulation, the whole being finally enclosed with sheet aluminium covers. It will be seen that seam welding provides an ideal way of building the inner member of such a unit.

Materials, Plant and Technical Control.

This paper would be incomplete without some remarks on the materials used and the table, Fig. 16, gives some of the properties of these alloys as they affect resistance welding. The oxide film present on the nickel-chromium alloys can be very troublesome and

Physical and Mechanical Properties of Materials, Fully Annealed, Which Affect Welding

Material	*Specification*	*Melting point, ° C.*	*Specific resistance, microhms/-cm. cube (15° C.)*	*% conductivity of copper*	*Thermal conductivity, G.G.S. units at 15° C.*	*Coefficient of expansion*	*Ultimate tensile strength, tons/-sq. in.*	*Brinell hardness*
Mild steel	0.1–0.15% carbon	1450	11.5	15	0.14	0.000012	20–25	90–110
Austenitic 18–8 stainless steel	18% chromium, 8% nickel, titanium or other elements present	1450	73	2.36	0.038	0.000018	35–45	160–200
Inconel	80% nickel, 14% chromium and 6% iron	1395	98	1.76	0.036	0.000014	35–42	140–180
Nimonic 75, 80–20 nickel-chromium alloy	80% nickel, 20% chromium and some titanium	1375	110	1.57	0.030	0.000015	42–50	160–200
H. R. Crown Max. centl. casting	23% chromium, 1% Si, 12% nickel and 3% tungsten	1410	87	1.98	0.030	0.000017	40–45	200–230

Fig. 16 Table of properties of materials

strict measures have to be taken to remove any oxides after heat treatment, prior to welding. The high electrical resistivity of these alloys also has to be catered for, particularly when seam welding the thicker gages. Another important point is the high coefficient of expansion which can give rise to considerable distortion when seam welding. Fortunately, all the alloys being corrosion resistant, they can be welded under water and advantage is taken of this when stitch or seam welding by directing a flow of water over the electrodes and the work, thereby reducing distortion and minimizing electrode pickup.

With regard to welding plant, all the welding machines in use are of the straight alternating-current transformer type and all are fitted with weld-time control. For tacking spot welds prior to stitch or seam welding it is only necessary that the spot weld should be strong enough to withstand the stresses set up during the subsequent welding operation. Foot-operated machines are used as being convenient in operation. Where the spot welds are structurally loaded, fully automatic air-operated machines are used. For stitch welding, mechanically operated machines are in general use, all being equipped with full electronic tube controllers, with phase-shift heat control. Naturally, all the seam-welding machines are fully electronically controlled and those machines on which the most critical welds are made are fitted with the latest constant current control which can take care of considerable fluctuations of voltage in the main lines.

The question of welding technical control is a vital one in view of the fact that a faulty weld might give rise to serious trouble on an engine and a careful check is kept on all the resistance-welding processes in use. This is done by presenting initial test pieces, conforming as near as possible to the weld in question, to the metallurgical laboratory for approval. This check generally takes the form of a radiographic examination for weld continuity and porosity, a tensile test for weld strength and a metallographic examination for fusion and metallurgical structure. All the necessary data is logged and, when approved, a weld instruction sheet is issued carrying all the relevant technical data such as machine called for, electrode thrust, weld time, primary current and transformer tapping, etc. Thereafter, tensile-test pieces have to be prepared prior to any particular weld being undertaken and these test pieces have to break above a given load as stipulated on the instruction sheet.

It is, unfortunately, not possible to cover the subject of welding technique, but it will be realized that a considerable amount of pioneer work had to be undertaken before satisfactory results were obtained. As time went on, experience showed that some of the more difficult welds could be eliminated by changes of design and the examples given show how an intelligent grasp of the fundamentals required for resistance-welding construction by the designers can result in a practical production job of what might otherwise have been a difficult project to make.

APPLICATION OF STATISTICS TO THE SCIENCE OF MANAGEMENT.

ACKNOWLEDGMENT.

This article by MR. J. M. JURAH of New York University is reprinted from "Mechanical Engineering" for April, 1949.

The Scientific Method.

Among the very great achievements of human progress has been the development of the scientific method. In essence, the scientific method involves the process of reaching conclusions from reproducible experiment, and not from opinion, hunch, mysticism, folklore, or other forms of guesswork. In detail, the scientific method follows a wellknown cycle of events:

1 We state a hypothesis.

2 We design an experiment to test the hypothesis.

3 We collect data.

4 We analyze the data.

5 We draw conclusions from the data.

6 We affirm, deny, or modify the hypothesis. We thereupon state a further hypothesis, and the cycle starts all over again.

The science of statistics plays an important role in the use of the scientific method. The design of experiments, the collection and analysis of data, and the concepts of error or significance, all rely on the methods of statistics for their validity. While the basis decision to use facts at all (rather than to use guesswork) is the truly fundamental achievement, the development of the science of statistics to collect and interpret facts is a most important satellite achievement.

Evolved in the crucible of the scientific method, many longstanding arts emerged as sciences. Alchemy emerged as chemistry, astrology as astronomy, mysticism as medical science. Against the measured beat of the great clock of human history, these changes have been revolutionary in their speed and in the violence of their impact.

Modern industry itself has emerged from the same crucible. Without the scientific method the monster mechanical tools which are the skeleton of our industrial system could not have been designed and built. Without the scientific method, the modern transport and communication necessary to feed these monsters could not have evolved. Without the scientific method, the mechanical energy required to breathe life into these tools could not have been harnessed.

Management by Guesswork.

Many managers of today are shocked to be told that business is managed by a methodology which has much in common with the methodology of the Middle Ages. Yet hunch, guess, suspicion, and solidified habit are the rule rather than the exception in management methodology. A few recent examples will be cited:

(*a*) In a plant making precision instruments, the designers, machine-tool

builders, and production officials had for decades debated the difficulties of meeting the tolerances on the tiny shafts turned by a large battery of precision screw machines. All except the designers agreed the tolerances were too close. All except the machine-tool builders agreed the machines were inadequate for the job. All except the production officials agreed the operators were careless. The solution had been for decades to have the inspectors sort the parts, a costly operation in itself, while enduring a greater additional cost due to losing more than 10 per cent of production because of defectives.

Then someone applied the scientific method. He measured, with great precision, 500 successive pieces turned out by one of the machines. Analysis of these simple data disclosed the following facts:

1 The machine was easily able to hold the tolerances.

2 The shop gages were inadequate to permit the operators to regulate the machines.

3 The rate of tool wear was so small that the machine, once properly adjusted, could go on for days producing perfect pieces.

4 Numerous suspicions of causes of variation were unfounded.

5 It was possible to do away with the sorting inspection.

6 It was possible to increase production substantially, both in good and in total pieces.

(*b*) A company making and selling building materials was experiencing great difficulty in meeting the prices of its competitors. There were the usual recriminations between the major deparments of the company. Then someone applied the scientific method. He analyzed the sales data and found that 5 per cent of the designs of product sold accounted for two thirds of the company's sales. He found that one half of the designs accounted for less than 5 per cent of the sales. A drastic reduction was made in the number of designs. The reduction in sales was small, but the reduction in costs was substantial.

(*c*) A company making chemical products found its maintenance costs high. The number of maintenance men virtually equaled the number of operators. Use of the scientific method disclosed that the ten most important types of equipment failures accounted for over 50 per cent of the maintenance costs. As a result of this disclosure, the company assigned engineering talent to redesign the equipment to eliminate failures of these types, with resultant substantial reduction in maintenance costs.

These examples are sufficient to illustrate what is realized by many, namely, that modern industrial operation is the source of many facts which can be sought out and analyzed to the great benefit of industry.

Acceptance of the Scientific Method in Management.

The author does not mean to imply that the managers of industry are blind to these potentialities. In some areas, the scientific method has taken firm root. The techniques of time-and-motion study, conceived by Taylor, Gantt, and the Gilbreths, have survived the era of the efficiency experts and have become part of the daily life of industries. Cost accounting has made great strides in measuring and predicting costs. There has been some acceptance of the aptitude test, the systems of merit rating, and other tools of personnel management which rely greatly on statistical methodology for their validity.

In the field of wage determination, the indexes of cost of living and of prevailing wages have been written into numerous collective-bargaining contracts and escalator clauses. Market research is just commencing to use sampling of consumer opinions and re-

actions. Sales forecasters are beginning to use measures of purchasing power and other economic factors.

This list of applications is all to the good. Yet the fact is that we are applying only a fraction of what we know. The rate of adoption of these managerial tools is so slow as to pose the following question: If the scientific method is so widely applicable to solution of problems of management, how is it that the use of this method is so limited?

This paradox will be discussed by considering the problems of applying the statistical methodology in a field in which the author spent some early years—that of "quality control." Because these problems of application of quality control parallel so remarkably the experience recorded in application of other instances of the scientific method (such as time study, cost accounting, or market analysis), it will be possible to generalize even though the examples are drawn from this one field.

The Scientific Method in Quality Control.

Few problems in management are more naturally susceptible to scientific analysis than the problem of quality control. Inspectors and operators make myriads of measurements. The resultant data can disclose not only the nature of the product but also the nature of the process which made the product. Through these data

1 The process can be improved to make the product better in the first place.

2 The process can be regulated to perpetuate this improvement.

3 The specifications can be improved through knowledge of process capability.

4 Detail inspections can be eliminated.

5 Scrap, reworks, and other losses can be minimized.

6 Consumer complaints can be reduced.

7 Quality reputation can be improved.

There is nothing theoretical in the foregoing. Personally the author has had experience with instance after instance involving many products and processes in many companies in which all of these things have been done, to the amazement and gratification of the managers involved.

Some of the vital techniques for doing all this are over 20 years old; accounts of the pioneering work done by the Bell Telephone Laboratories and Western Electric Company were published before World War II. During World War II the problems of expanding mass production of precision apparatus fairly cried for improved means of quality control. Publicity was given to the statistical tools. Training courses were held. Many of the men trained became enthusiasts for the new techniques. The enthusiasm has given birth to an American Society for Quality Control, to a rash of literature, and to a series of frustrations for most of the enthusiasts who have endeavored to translate the techniques into action. The proved techniques have made comparatively little headway.

The failure to apply the techniques has certainly not been due to any desire on the part of managers to perpetuate scrap losses, inspection costs, and customer complaints. The managers quite obviously want to improve their performance in these respects. Granted, there is some indifference among those managers who are drugged by favorable profit statements and who are not driven by economic necessity to spend time on how to improve operations. But in the main, managerial attitude toward the objectives of quality control is one of longstanding agreement.

Neither has the failure to apply the techniques been due to lack of technical

knowledge on the part of the industrial statisticians. In the author's experience, the bulk of the quality-control problems are, in statistical contemplation, rudimentary rather than complex. Even an amateur statistician can solve most of them; so much so, that some managers and engineers constantly solve these problems without realizing they are making repeated sojourns into the field of statistics. They are much like the man who speaks prose without knowing it.

We must grant that some of the present "quality-control engineers" are of a very recent vintage. However, they know enough of statistical techniques to solve the bulk of the statistical problems which they encounter. Their failure lies not in their lack of statistical knowledge. It lies in their lack of knowledge of the nature of the managerial process and, in particular, in their lack of knowledge of human relations.

Starting Big versus Starting Small.

Here are a few actual (though disguised) examples. The W. Company is a large manufacturer of electrical apparatus. The vice-president of manufacture became convinced that quality control by statistical methods had broad application to his work. Providing a generous budget he promptly set up a quality-control department of about 20 engineers. In short order, the plant superintendents and foremen found themselves confronted with men who presented mysterious charts with data on defects, who wanted to investigate the processes, who proposed many innovations to help these superintendents who wanted no charts, no investigations, no innovations, and no help. The inevitable consequence was a full-scale jurisdictional deadlock. At the end of a year the vice-president reviewed accomplishments and found that nothing had been accomplished. Thereupon he disbanded the quality-control department.

In contrast is the experience of the Y. Company. Here the vice-president was also convinced that modern quality-control methods could improve his operations. However, he engaged only one engineer to apply the methods. Furthermore, because no one in the company had ever installed a system of modern quality control anywhere, the vice-president engaged a qualified management consultant, one who had supervised many such installations, to help the engineer chart a course. For the first few months the engineer was strictly confined to one operating department. The subsequent results were beyond all expectation in reducing scrap losses and cost of inspection.

As a consequence of this success in the first department, it was decided to expand the quality-control work to other departments. Consideration was given to holding a meeting of all foremen to explain the gratifying results obtained and to have the foremen of the first department "testify" to his fellow foremen that there was nothing to fear. It was then found that no such meeting would be necessary. The story had already been told by the "grapevine." The foremen were already "sold" and were competing with each other for the right to be the next to obtain the services of the quality-control engineer.

The decisive factor which contrasts these two experiences is that, in the successful instance, it was recognized that time is an essential ingredient in the solution. Adoption of the scientific method is a fundamental change requiring time to consummate. The larger the mass to be changed, the greater the time necessary. The broad front attack in the W. Company represented too violent a rate of change and was doomed to failure from the outset. The idea of a guided gradual growth, as adopted by the Y. Company, greatly increased the odds for success.

The Need to Sell the Techniques.

To many quality-control engineers, the inability to "sell" the shop executive is the main cause of frustration. From the viewpoint of the quality-control engineer, here is a proved set of techniques whose logic is unassailable; "therefore" anyone who does not accept the logic is an obstructionist and should be commanded to comply. This is all very well, but it ignores the views, the responsibilities, and the rights of the shop executive.

Sound organization practice requires that if the shop executive is to be held responsible for conduct of shop affairs, he must be given coextensive authority to regulate those affairs. If the top management adheres to this sound organization practice, it will not tell the shop executive he must adopt certain techniques. To tell him he "must" do something is to relieve him of the responsibility for any failures which might be assigned by him, rightly or wrongly, to these techniques. The top management can only ask that the shop executive give consideration to these techniques. From there on, the quality-control engineer is on his own. He must sell his service by demonstration, by persuasion, by patience, and by all other known devices of salesmanship, including of course the excellence of his service.

To most engineers or scientists, selling seems to be an unnecessary, distasteful, and degrading chore. That they think so is an indictment of their technical training quite as much as an indictment of their lack of experience in human relations. Actually, there is no escape from the need to sell, nor should there be. If the quality-control engineer cannot convince the shop executive through the democratic process of free choice the matter should end there. Progress achieved through dictatorial methods is achieved at too high a price in human relations.

The Methods of Selling.

On the face of it, the requirement that the techniques must be sold rather than forced on the shop executive means that there will be no progress. The precise opposite is true—only through genuine conviction is there any enduring progress. As stated previously, the shop executive for many years has been in sympathy with the objectives of quality-control. Yet the usual approach of the quality-control engineer is to dwell on these objectives as though they were unknown to the shop executive. As to these objectives, it is the quality-control engineer who has newly rediscovered them, not the shop executive.

In contrast, the quality-control engineer devotes little time to explaining, in shop language, how the new methods are to be any improvement over traditional methods for achieving these long-standing objectives. The common, and fatal, approach is to try to teach mathematical statistics to the shop executive.

Not long ago, after the author had just gone through explaining to a vice-president what was involved in adopting modern quality-control, he confided to me that he had once sent some executives to a wartime course in quality control to find out what it was all about. They came back with the report, "quality control is a blackboard full of higher mathematics," He was unable to see the connections between that report and his work, so he dropped the whole thing.

The shop executive has some honest doubts as to how an engineer who has never seen a particular process can arrive at a knowledge greater than that possessed by supervisors who have been associated with that process for 25 years. These doubts cannot be answered by paternal smiles; they must be answered squarely. In the case of the precision screw machines related

earlier in the paper, a comparative handful of data ended an argument which had gone on for decades. The ability of the engineer to develop the new knowledge was purely the result of his being armed with a new tool—the tool of the scientific method—the use of facts rather than debate to arrive at conclusions.

The author often has used analogy to explain this. A shop supervisor has for 25 years been inspecting surfaces with the naked eye. One day an engineer invades the department armed with a microscope. On that day, the engineer, with only one day's observation, can make new disclosures about those surfaces to the man who has examined them for 25 years. Or, consider the fast-moving mechanism which is photographed by a high-speed camera. On the day the film is projected, there are available new disclosures to the man who has worked with that mechanism for 25 years.

In all three instances, it was the new tools which made the new disclosures possible. The analysis of data, the microscope, the high-speed camera are of the same species. The last two, being physical, tangible devices, are more obvious than the "mental" tool of analysis of data. But all three can be explained to the shop supervisor as tools.

Laboratory Tools and Shop Tools.

A common cause of failure is the inability to distinguish between laboratory tools and shop tools. The ordinary micrometer is less than a hundred years old. We can easily visualize that in the early days of machine-shop practice, a measurement by micrometer was quite a ritual. The shopman went to the laboratory and waited respectfully while the engineer (in a laboratory coat) took the instrument from its velvet case and made the measurement. Over the years, the instrument has been simplified, and shop training broadened to the extent that today the micrometer is in widespread use as a shop tool.

The use of statistical tools will follow a similar pattern. Some of these tools are already so simple that they can be put into shop hands at once. It is often very useful to chart measurements of individual units of product against the design tolerance limits. From his experience, the author has yet to encounter an operator or shop supervisor who could not understand such a chart, and therefore this chart is classified as a shop tool.

On the other hand, it takes me several class sessions to get across even to graduate engineering students the concept of the Shewhart control chart for comparing averages with statistical boundaries for variations in averages. The Shewhart chart, along with problems in analysis of variance, theory of sampling, and a host of others, is in the laboratory-tool category. Any effort to force these tools prematurely on the shop is bound to generate the resistance which accompanies suspicion of the unknown.

It is not meant that the engineer is precluded from using these laboratory tools. The lesson is merely that laboratory tools are to remain in the laboratory until it is timely for them to go into the shop.

The Need for a Program.

Finally, the quality-control engineer, in the absence of broad managerial experience, does not realize that introducing a new methodology requires a program or series of events. The actual use of the new techniques is only one of these events.

The breadth to which this program can extend is exemplified by the actual program of a company in the textile industry. This program includes the following:

(*a*) Economic analysis of what is to be gained by achieving improved quality.

(*b*) Measurement of competitors' quality and comparison with the company's own quality.

(*c*) Raising the level of quality-mindedness among all levels of personnel.

(*d*) Improved organization of the inspection department.

(*e*) Improvement in completeness and adequacy of quality specifications.

(*f*) Development of adequate check inspection and quality audit procedures.

(*g*) Measurement of individual operator quality performance.

(*h*) Improvement of materials purchased from vendors.

(*i*) Development of training programs for various classes of personnel.

(*j*) Identification of major projects for quality-control research.

(*k*) Establishment of steering committees for achieving co-ordination among the principal departments of the company.

In the framework of such a list, the detailed application of the statistical techniques is seen in correct prospective. The statistical techniques are never a program in themselves. They are only part of a broader program. Failing to realize this role of the techniques, the quality-control engineer can fail for lack of perspective quite aside from other considerations.

Conclusions.

We can generalize from the example of quality control to draw conclusions which apply to the use of statistical methods in the science of management.

(*a*) Statistics is not an end in itself but only an adjunct in applying the scientific method.

(*b*) The practitioner of statistical methods must be aware of the broader program of managerial endeavor of which his specialty is only one element.

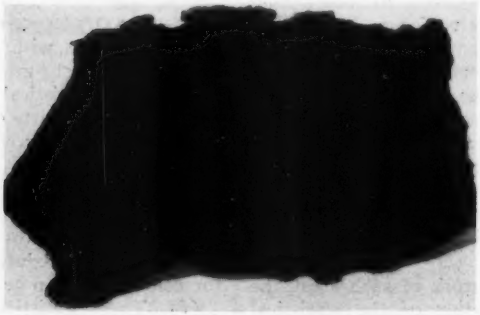
(*c*) Application of statistical methods must commence on a small scale and must grow with due regard for the time needed to grow solidly.

(*d*) The statistical methods must be sold by all the arts of salesmanship. They cannot be forced on the executives of industry.

(*e*) The intricate methodology must be kept in the background for use by the engineer only; the simple methodology may be introduced into the shop for use by the shop personnel.

(*f*) The statistical practitioner must learn of the technology of management. He cannot expect that the managers will want to learn of the techniques of statistics; they will want to learn only how these techniques can aid them to carry out long-standing management objectives.

Primary Carbon Residue



High Vanadium Precoat Slag

Final Compound Slag

Stages of Slagging in Saturated Nest

Plate 1

DISCUSSION.

DISCUSSION OF THE PAPER "FIRESIDE DEPOSITS—A STUDY OF MINERALS CONTAINED IN FIRESIDE DEPOSITS OF OIL FIRED BOILERS, BY W. A. LAMBERTSON."

By Frank E. Clarke.*

The author erroneously attributed the theory of vanadium precoat slagging to Hock (1), and concluded that the major crystalline phases of oil furnace slag have been identified as sodium sulphates. Actually, the vanadium theory was published by the writer in a discussion of the Hock paper in August 1946 (2). Continuous investigation of slags since that time has strengthened the writer's belief that vanadium compounds play an important part in slagging. The basic observations of this investigation are summarized briefly in this discussion as evidence that the most pertinent constituents of oil furnace slag still may not have been identified.

The following generalizations are applicable to the various specific slag problems investigated by the writer.

A. Slagging starts with relatively slow development of a thin, adhesive pre-coat slag.

B. Vanadium compounds generally are the principal constituents of the pre-coat.

C. Vanadium pre-coat slag instigates rapid development of massive sodium sulphate slag.

The relatively low melting points and excellent fluxing properties of vanadium compounds make it possible for them to wet boiler tube surfaces over a wide range of operating conditions. In high temperature superheaters, they are believed to deposit by the "chemical icing" mechanism suggested in (2). This assumes that molten droplets adhere to surfaces whose temperatures approach the fusion temperature of the droplets, just as rain ices telephone wires. Superheater pre-coat slag thus formed contains approximately 50% by weight of vanadium compounds. Heavy sodium sulphate slagging appears to start after the pre-coat attains a thickness of 1/32″ to 1/16″. Careful examination of superheater slag fragments generally will disclose the thin pre-coat shell on the tube side of the deposit; however, it is possible for certain areas to be deslagged by thermal shock and subsequently covered by drifting sodium sulphate slag.

An almost identical pre-coat slag will develop in low temperature generating nests if atomization and combustion are faulty. In this case, impinging oil wets the tube surfaces, burns in place and vanadium compounds flux the ash into a pre-coat slag. Combustion difficulties with newly installed, 2000 psi oil fired boilers afforded this Station an excellent opportunity to follow the metamorphosis of this vanadium pre-coat slagging in generating nests. Repeated trials showed that short period operation

* Chemical Laboratory, USN Engineering Experiment Station, Annapolis, Md. The opinions expressed in this discussion are those of the writer and do not necessarily reflect the views of the Navy Department nor of the Naval Service at large.

(up to 500 hours) on Grade 2 boiler fuel deposited a streaky, shiny, black film which resembled dried oil. This contained approximately 70% by weight of carbon, and vanadium compounds were present in the inorganic fraction. Continued operation developed this primary film into a 1/16" compact, black, crystalline, pre-coat slag, in approximately 1000 hours. After this period of operation, heavy stucco-like grayish brown slag built rapidly on the pre-coat. Plate 1 shows the appearance of the primary carbon residue, the vanadium pre-coat slag and the final compound slag. The arrow indicates the black pre-coat slag on the tube side of the final deposit. The analyses of representative samples in Table I show the remarkable similarity of pre-coat slags from superheater and generating nests. This uniformity of composition is further confirmed by the fact that the X-Ray pattern shown in Column A of Table II is obtained regardless of whether the pre-coat forms in the superheater of a Naval boiler or the generating nest of a stationary boiler. Incidentally this same fact is true of the final massive slag. The X-Ray pattern of the pre-coat slag has not been positively identified but is believed to be sodium vanadyl sulfate.

If, as these observations would appear to indicate, vanadium is the instigator of slagging, it is of little importance that sodium sulphates constitute the major portion of the final deposit. The writer feels that the author's attempt to identify the minerals contained in fireside deposits is a promising approach to proper understanding of the mechanism of slagging. It is hoped that these studies will not be concluded until the vanadium compounds related to the problem have been satisfactorily identified.

TABLE I.

COMPOSITION OF PRE-COAT SLAGS.

Apparent Constituent	Percent		
	Generating Nests		Superheater Nest
	A	B	
Vanadium Oxide, V_2O_5	61.57	66.51	67.75
Sodium Sulfate, Na_2SO_4	30.61	22.08	30.61
Nickel Sulfate, $Ni\ SO_4$	—	3.43	—
Nickel Oxide, NiO	—	2.07	—
Iron Oxide, Fe_2O_3	1.90	—	1.80
Magnesium Sulfate, $Mg\ SO_4$	—	4.48	—
Calcium Sulfate, $Ca\ SO_4$	—	1.21	—
Silica, SiO_2	0.18	0.50	0.18
Moisture	2.54	—	—
Organic	2.22	0.65	0.30

BIBLIOGRAPHY

(1) JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Volume 57, No. 4, pp. 508 (1945).

(2) JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Volume 58, No. 3. pp. 467-469 (1946).

TABLE II.

X-RAY PATTERNS OF PRE-COAT SLAG AND MASSIVE SLAG.

A * Pre-coat Slag		B Massive Slag		C Sodium Sulfate, Na_2SO_4	
d	I	d	I	d	I
9.66	.1	5.18	.3		
7.40	.2	4.67	.8	4.66	.4
7.08	.1	4.26	.1		
4.79	.1	4.04	.2		
4.68	.1	3.93	.1		
3.86	.1	3.84	.1	3.84	.13
3.74	.2	3.52	.2		
3.56	.1	3.43	.2		
3.40	1.0	3.17	.7	3.18	.33
3.23	.5	3.11	.1		
3.08	.7	3.07	.7	3.08	.27
3.02	.1	2.97	.1		
2.93	.6	2.90	.2		
2.79	.5	2.84	.1		
2.73	.3	2.78	1.0	2.78	1.0
2.65	.2	2.64	.6	2.64	.4
2.53	.1	2.57	.1		
2.45	.1	2.51	.1		
2.38	.1	2.38	.1		
2.33	.1	2.32	.5	2.33	.27
2.26	.2	2.20	.1		
2.19	.7	2.05	.3		
2.06	.1	1.96	.1		
1.97	.2	1.92	.1		
1.87	.3	1.89	.1		
1.80	.7	1.86	.9	1.87	.53
1.75	.2	1.83	.2		
1.72	.1	1.797	.2	1.79	.05
1.68	.1	1.712	.2		
1.65	.1	1.677	.4	1.67	.7
1.604	.1	1.660	.3		
1.581	.1	1.602	.3	1.60	.05
1.557	.4	1.579	.2		
1.536	.2	1.548	.7	1.55	.2
1.494	.3	1.527	.1		
1.463	.1	1.508	.1	1.50	.08
1.432	.1	1.491	.2		
1.391	.3	1.479	.1		
1.333	.2	1.467	.1		
1.297	.2	1.452	.1		
1.275	.2	1.424	.3	1.428	.11
1.254	.1	1.408	.1		
1.229	.2	1.382	.2	1.385	.05
1.192	.2	1.372	.1		
1.172	.1	1.318	.3	1.320	.04
1.149	.2	1.300	.1		
1.089	.2	1.291	.4	1.295	.16
		1.275	.4	1.280	.08
		1.252	.1		
		1.229	.1		

* An identical pattern was obtained on a compound synthesized from Na_2SO_4, V_2O_5 and H_2SO_4, presumably $NaVSO_4.H_2O$.

BOOK REVIEWS.

THE A.R.R.L. ANTENNA BOOK (Fifth Edition—1949) was written by the Headquarters Staff of the American Radio Relay League. The book is paper bound and consists of 288 pages measuring 6½ inches by 9½ inches. There are 831 illustrations including 72 charts and tables in addition to 72 basic formulas. The price is $1.00 in the United States, its possessions and Canada. Elsewhere the price is $1.25.

REVIEWED BY: COMDR. CHARLES W. HARRISON, JR., U.S.N.

The 1949 edition of the A.R.R.L. Antenna Book is divided into two principal parts. The first portion of the book is devoted to a discussion of some of the fundamental aspects of antennas, transmission lines and wave propagation. The second portion covers specific antenna designs for the various amateur frequencies, including the highly important mechanical features of construction. Related subjects, such as determining geographical directions, are discussed.

The first five chapters of the book are devoted to underlying principles of antennas and lines. An easy to understand qualitative introduction to electromagnetic theory is given. A very intelligent exposition is presented regarding coupling of antennas and transmission lines, unbalanced transmission line currents, etc., not normally found even in the better radio engineering texts. This book proves that advanced concepts can be presented simply and yet correctly at the same time! Many of the available simple electronics texts are a comedy of errors.

In the interest of technical accuracy the reviewer would like to suggest a few changes in the "theoretical" portion of the book. These might be incorporated in the next edition. It is to be emphasized that they do not invalidate the intended purpose of the book; that of providing practical design information relating to antennas and transmission lines.

Figure 1-1 page 9 represents a plane wave traveling along the ground. The electric and magnetic fields are labeled "electrostatic lines of force" and "magnetic lines of force," respectively. It should be noted that electrostatics and magnetostatics play no part in the theory of radiation from antennas nor in the propagation of electromagnetic waves. There is nothing "static" about it! On page 11 is a paragraph entitled "Field Intensity" some confusion arises because no distinction is made between field intensity and field strength. Usually field intensity is considered to be proportional to the square of the field strength. If this definition is used for field intensity the paragraph entitled "Attenuation" on page 11 might have, as an opening sentence, "In free space the field intensity of the wave is inversely proportional to the square of the distance from the source."

On page 25, first column, a distinction should be made between a self-resonant antenna and an antenna that is tuned to resonance. There is nothing particularly magical about a self-resonant antenna or a self-antiresonant antenna. If the resistive component of the input impedance of an antenna is large compared to the ohmic resistance of the circuit used to tune out the reactance, the antenna will perform efficiently as a radiator, provided earth losses are small. At the beginning of the second column, page 25, it might be well to delete all reference to "distributed constants" of an antenna. These concepts are of convenience in analyzing transmission lines, not antennas.

It is suggested that all mention of voltage distribution along an antenna be dropped from the book. One can measure the charge distribution along an antenna. If voltage distribution is to be measured, where is the reference for voltage in a circuit which is normally large compared to the wavelength?

The section on antenna impedance beginning on page 28 is good. The proposition of impedance of an antenna at any point along the wire is de-emphasized—and rightfully so. If the currents flowing out of one terminal and into an adjacent terminal are not equal, one is in a quandary to define antenna impedance.

On page 36 under the heading "Pick-Up Efficiency" the concept of effective area of a receiving antenna is introduced. This is unfortunate because for each value of load impedance it is necessary to define a different value of effective cross-section for the very same receiving antenna!

It is surprising to discover in this practical book on antennas how many fine points relating to the subject are discussed. For instance it is pointed out that the current is not zero at the ends of an antenna because current must flow radially inward to charge the end faces of the wire!

A little over 70 pages of the book are devoted to transmission lines. Considering this fact perhaps a more descriptive title for the book is in order. A suggested one is "The A.R.R.L. Antenna and Transmission Line Book." The transmission line section contains a wealth of information of practical value, from impedance matching to construction of bazookas. Inasmuch as the discussion is based on dissipationless transmission lines it is believed that increased clarity would result by dropping the term "characteristic impedance" and substituting "characteristic resistance" of a line. A line is then non-resonant when terminated in a resistance equal to its characteristic resistance and any anxiety caused by the knowledge that a phase angle is ordinarily involved in an impedance is avoided. For example, the characteristic impedance of an air-insulated parallel-conductor line is given on page 81 as

$$Z_o = 276 \log \frac{b}{a}$$

This is obviously not a complex expression and therefore is not an impedance in the conventional sense. The expression acquires its dimension of resistance because the number 276 is dimensionally a resistance.

The contents of this book if carefully read and studied should prove extremely useful to antenna design engineers, amateurs, and short wave enthusiasts. Some of the antennas described would be of interest even to the "better-half"—for emergency clothesline applications!

SCAVENGING OF TWO-STROKE CYCLE DIESEL ENGINES by A. H. Schweitzer. Published by The MacMillan Co., 60 Fifth Ave., New York City.

REVIEWED BY: DR. LASKAR WECHSLER.

Information on the scavenging of two-stroke cycle Diesel engines has hithertofore been scattered through numerous publications and text books. In his book "Scavenging of Two-Stroke Diesel Engines," Dr. Schweitzer has collected this information, added to it, and produced a single source of valuable data on design and testing of scavenging systems for two-stroke Diesels. A condensed version of this book was published in Diesel Power and Diesel Transportation in 1942 and 1943. Expanded and brought up to date, the same material is now presented as a book.

Before going into the detailed analysis and calculations pertaining to the design of the scavenging systems for two-stroke engines, the author very wisely devotes one chapter to the definition and explanation of all terms and symbols used in the text. Basic formulae are derived and their use illustrated. For convenience a table of nomenclature is printed on a fold-out sheet at the end of the book, so that it may be referred to continuously as the text is studied.

Next formulae and charts are presented to show the effect of scavenging efficiency and air quantity on brake mean effective pressure. This is followed by a description of the various types of scavenging systems. Charts are included to show the scavenging efficiency which may be expected from each system at various air delivery rates and scavenging pressures. A description of types of porting and their general design limitations is followed by the detailed procedures for the design of inlet and exhaust ports. This covers such vital information as, determination of port dimensions, delivery ratios, scavenging pressures, port opening and closing times, exhaust lead, and methods of supercharging. Design charts are included which greatly simplify calculations and selection of optimum designs. Lengthy derivations are included in appendices rather than in the text to simplify reading.

Inasmuch as the design of the blower, intake and exhaust piping, and receivers have a marked effect on the scavenging and performance of two-stroke cycle engines, several chapters are devoted to performance of various types of blowers, selection of receiver and expansion chamber volumes, exhaust pipe area and length. Methods of improving performance by tuning of exhaust systems and by eliminating interference are discussed. One chapter deals exclusively with the Kadenacy system of scavenging.

Numerous examples are worked out in the text to illustrate the use of the design charts. In addition a complete porting calculation is included to integrate the whole calculation procedure.

After a discussion of highly supercharged two-cycle engines, the author treats with the experimental phases of scavenging system design. Methods of testing for trapping efficiency, scavenging efficiency, exhaust lead, closure of inlet ports, balance of inlet and exhaust port areas, and exhaust and intake pipe tuning are discussed. The instrumentation required for special tests is illustrated. Means of correcting deficiencies and improving performance are described.

A tabulation of the scavenging system design data of 31 foreign, domestic, production and experimental Diesel engines is included. An excellent bibliography, including many references to foreign publications is appended and much information from these references is included in the text. "Scavenging of Two-Stroke Cycle Diesel Engines" is clearly written and profusely illustrated, and should be a useful addition to the library of any engineer engaged in the design, testing or improvement of two-cycle Diesel engines.

DEATH NOTICE.

The Society has learned with regret of the death in an airplane accident, of Mr. Charles Hosmer Morse, III Vice-President in Charge of Manufacturing, Fairbanks, Morse & Co., Chicago, Illinois.

A Civil Member.

CHANGES IN MEMBERSHIP.

Since the publication of the May, 1949, JOURNAL, the following changes in membership in the Society have occurred.

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Buchanan, Myron W.

ASSOCIATION NOTES.

Original Articles.

The editor of the JOURNAL is most anxious to obtain manuscripts of original articles on subjects of interest to naval engineers. Any manuscript accepted for publication becomes the property of the Society which copyrights it upon publication. Authors are paid from $50.00 to $150.00 depending upon length, interest and professional value

The rules for manuscripts are as simple as we can make them.

1. They must be legible (typewritten, double-spaced preferred) since no proof is submitted to authors for correction prior to printing.

2. Single copy only is required.

3. Mathematical formulae used must be set up, preferably by hand printing, so that there can be no question as to symbols, subscripts and exponents.

4. Any drawing submitted must be black on white. A cloth or paper tracing is acceptable but not required.

5. Photographs may be negatives, but glossy prints are acceptable.

6. Include on a separate page a short biographical sketch(es) of the author(s). 50 to 100 words for each author is desired.

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American Society of Naval Engineers
Bureau of Ships, Navy Department
Washington 25, D. C.

Manuscripts accepted will not be returned unless specifically requested by the author. If returned, they will be in the condition which has resulted from the work of the printer and the engraver. Immediately following publication, the author is furnished 10 reprint copies of his article free of charge. Additional copies may be purchased from the Society if the request is received 30 days prior to the publication date which is the 25th of the issue month. Estimate of cost of additional reprints, which will vary with the nature of the article and the number of copies ordered, will be furnished on request as soon as possible after the article is set up.

Manuscripts not accepted will be returned, postpaid, by the Society.

Manuscripts which are accepted will be published in the earliest JOURNAL which has not yet been closed (60 days before publication) and for which insufficient material is already on hand.

Subject Matter and Authors.

To assist the editor in programming future issues, the help of all members is solicited. To make it easy, the last page in this issue is readily detachable as you will find. This page has blanks for two entries.

(*a*) What subject would you, as a member, like to see covered by an article in the JOURNAL.

and

(*b*) Who do you consider best qualified to prepare such an article authoritatively, and what is his address. (if you know).

Any of these forms which are filled in and mailed to the Society will be received with thanks and the editor will follow up to get the suggested articles prepared.

PRINTING.

Each reader must be his own judge as to the comparative merits of any two issues of the JOURNAL. The quality of material is a composite measure of the value of the manuscript received and the Editors' selection of these with reader interest in mind. Qualitatively we do the best we can.

It was announced in April that the format was to be changed for two purposes. First was improvement in readability. Since practically no comments have been received on this point, no conclusion will be drawn. Second was reduction in cost. Quantitatively the May JOURNAL was about 20 per cent larger than the February issue. The printing cost showed a reduction of 13 per cent. Mailing weight also was down 20 per cent.

Some experimentation is continued in this issue. Margins have been slightly increased resulting in a reduction of about 10 per cent in wordage for page.

The November issue will probably mark the final attempt at format variation before decision on approximate standardization is made.

Comments on this subject would be appreciated.

PHOTOGRAPHS.

Photographs of current or historical interest to readers of the Journal are desired' Any which are accepted by the Society will be purchased by the Society at a standard price of $5.00 each. Such purchase *will not* include any copyright by the Society, but the contributor must hold the Society free from any charge of violation of any previous copyright.

NOTICE OF DEATHS.

The Society has no satisfactory machinery for obtaining notice of deaths of members. It is particularly desired that we receive such notice upon the death of any past officer, past member of the Council or member of more than twenty years' standing with photographs and short obituaries. It will be appreciated greatly if any member who learns of the death of a member will advise the Secretary-Treasurer.

ADDRESSES.

Once more attention is invited to the necessity of keeping the Society advised as to changes in addresses.

SOCIETY LAPEL BUTTON.

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a very fine dignified insignia. It is one-half inch in diameter.

The oak leaves and lettering are red on a gold back-ground.
It is available to all members at fifty cents (50c) each.

Powerful Example of GM Diesel-Electric Drive

ANOTHER powerful demonstration of General Motors Diesel-Electric Drive is this picture of the 34,183-ton R.M.S. Caronia being eased into her berth against an ebb tide by six units of the famous "M" Diesel-Electric fleet of the Moran Towing & Transportation Company, Inc.

For docking liners at coastal ports, pushing barges, for cargo vessels, or wherever sure-fire flexible power in compact, weight-saving units is needed, GM Diesel-Electric Drive is first choice. What's more, it's economical with fuel, maintenance and upkeep. GM Diesel-Electric Drive is the power plant that pays its way!

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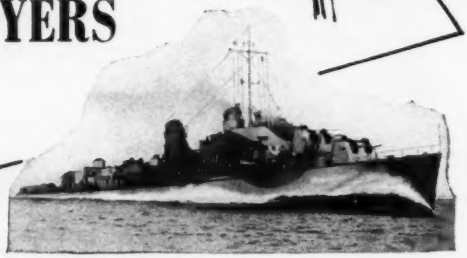

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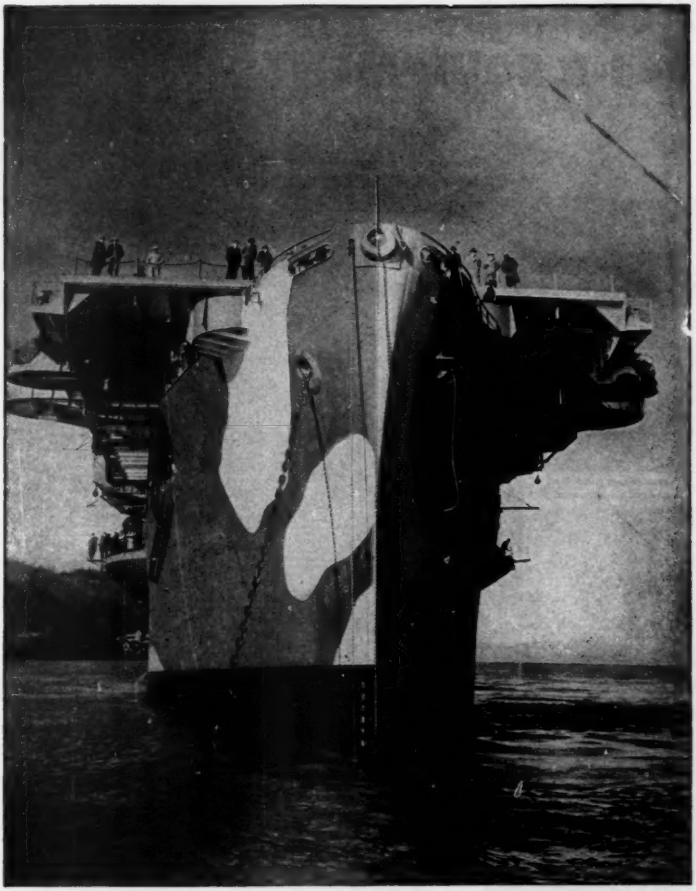

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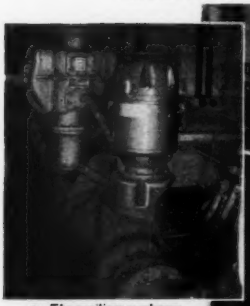
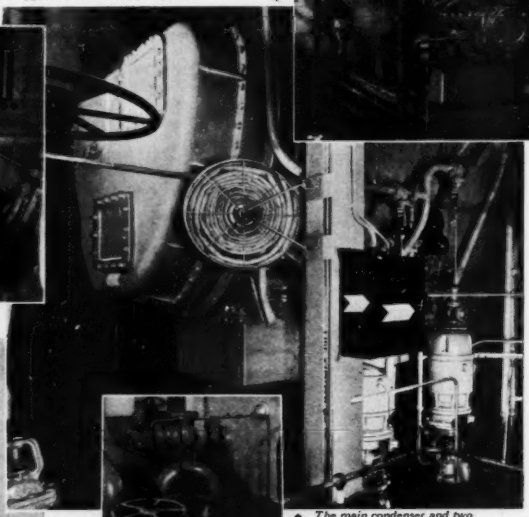


⬆ *The auxiliary condenser and circulating pumps.*



⬆ *The main condenser and two Class 2VHM condensate pumps.*

⬅ *One of three Class 6GTM cargo pumps on each vessel.*

⬆ *The two Class 2CRV sanitary and engine-room salt-water pumps.*

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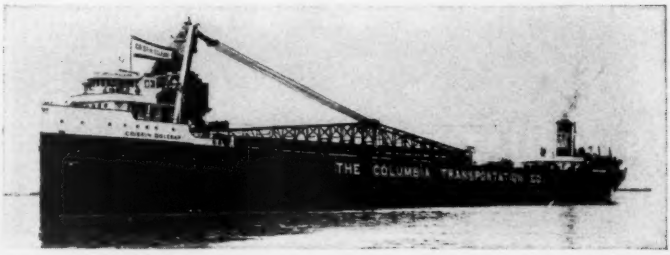

The Columbia Transportation Company's SS *Crispin Oglebay* has an all-electric conveyor system powered by General Electric turbines, motors and control which permits this vessel to put in at more ports, make quicker turn-arounds. The ship's 240-foot boom conveyor, longest ever used on a self-unloader, is almost half the ship's length.

In unloading, port and starboard hold conveyors bring limestone or coal to a forward elevator conveyor, which takes it up to a hopper at the pivot end of the boom conveyor. Conveyors are driven by four G-E induction motors totalling 750 hp. They help increase unloading speed from a normal 1800 long tons per hour—for ships of this type—to 2200. Power for all the unloading equipment is supplied by two G-E turbine-generators.

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Photograph courtesy of Cummins Engine Company, Inc., Columbus, Indiana

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Date.....

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I hereby apply for membership in the U. S. Naval Institute and enclose $2.00 in payment of dues for the first year to begin with the.....issue of the PROCEEDING (the monthly magazine of the U. S. Naval Institute).

I am interested in the objects and purposes of the Institute, namely, the advancement of professional, literary, and scientific knowledge in the Navy. I am a citizen of the United States and understand that members are liable for dues until the date of receipt of their written resignations.

Name.....

Address.....

.....

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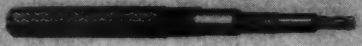

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(b) Multi-conductor Navy Type power cable

(c) Varnished cambric marine cable with basketweave armor

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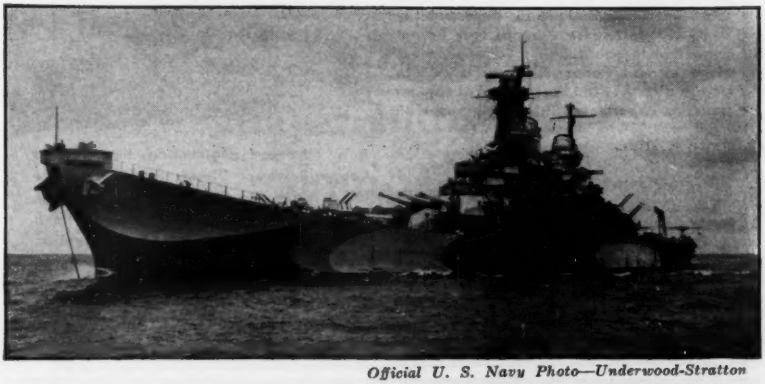

Official U. S. Navy Photo—Underwood-Stratton

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It was 50 years ago that the Electric Boat Company completed its first submarine, the Holland—the first undersea craft commissioned by the U. S. Navy. She was an experiment, regarded by many as of doubtful value.

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Official U. S. Navy Photograph

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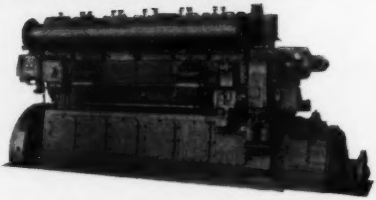

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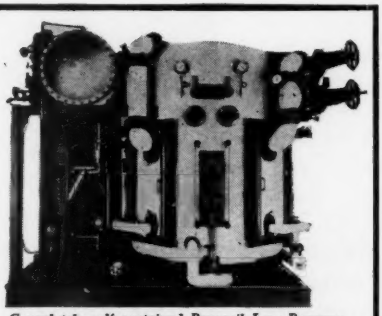

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.....

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Persons in civil life whose knowledge of engineering is such that they can cooperate with Naval engineers in the promotion of professional knowledge may be eligible as civil members. They shall have been in the active practice of an engineering profession for at least eight years and in responsible charge of important work for five years, and shall be qualified to design as well as to direct engineering work. Fulfilling the duties of a professor of engineering who is in charge of a department in a college or school of accepted standing shall be taken as an equivalent to an equal number of years of active practice. Graduation from a school of engineering of recognized standing shall be considered as equivalent to two years of active practice. Persons eligible as civil members may be admitted upon application and payment of annual dues, provided that the application is accompanied by the recommendation of two members and provided that the application shall receive the approval of a majority of the Council.

Persons in civil life who are not eligible for civil membership, but who are especially interested in naval matters or the merchant marine may be eligible as associate members. Commissioned officers of the United States Army and of foreign military and naval services may be eligible as associate members. Persons eligible to associate membership may be admitted upon application and payment of annual dues, provided the application have the recommendation of a member and provided the application shall receive the approval of a majority of the Council, except that in the case of commissioned officers of the United States Army and of foreign naval and military services, the recommendation of a member will not be required.

Associate members shall be entitled to all the privileges of other members except voting and holding office.

The annual dues shall be $5.00, payable on 1 January in advance.

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Name _____

Rank _____ File No. _____

Business connection and position, if any _____

For Civil Membership

(First) (Middle) (Last)

Name _____

Years in engineering work _____

Years in responsible charge of important work _____

Present business connection and position _____

Recommended by (two members) _____

_____

For Associate Membership

(First) (Middle) (Last)

Name _____

Rank, if Commissioned Officer of U. S. Army or of foreign military or Naval service _____

Business connection and position _____

Recommended by (one member) _____

_____

Signature of Applicant

Address for Journal and Mail

_____

_____

MAIL TO SECRETARY-TREASURER
THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.
BUREAU OF SHIPS, NAVY DEPT., WASHINGTON, D. C.

* See reverse side for required qualifications for various classes of membership.

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1894-95 P. A. Engineer R. S. GRIFFIN, U. S. Navy
1896-97 P. A. Engineer F. C. BIEG, U. S. Navy
1898 P. A. Engineer W. M. McFARLAND, U. S. Navy
1899 Chief Engineer A. B. WILLITS, U. S. Navy
1900 Lt. Comdr. A. B. WILLITS, U. S. Navy
1901 Lieutenant B. C. BRYAN, U. S. Navy
1902 Lieutenant C. W. DYSON, U. S. Navy
1903 Lt. Comdr. JOHN R. EDWARDS, U. S. Navy
1904 Lieutenant M. E. REED, U. S. Navy
1905 Lieutenant W. W. WHITE, U. S. Navy
1906 Lieutenant C. K. MALLORY, U. S. Navy
1907-08 Lt. Comdr. T. C. FENTON, U. S. Navy
1909-10 Lieutenant H. C. DINGER, U. S. Navy
1911 Commander U. T. HOLMES, U. S. Navy
1912 Lieutenant JOHN HALLIGAN, U. S. Navy / Lt. Comdr. E. L. BENNETT, U. S. Navy
1913 Lieutenant O. L. COX, U. S. Navy
1914 Lt. Comdr. H. C. DINGER, U. S. Navy
1915-16 Lieutenant A. T. CHURCH, U. S. Navy
1917 Lt. Comdr. J. O. RICHARDSON, U. S. Navy / Lt. Comdr. F. W. STERLING, U. S. Navy, Retired
1918 Lt. Comdr. F. W. STERLING, U. S. Navy, Retired
1919 Lt. Comdr. F. W. STERLING, U. S. Navy, Retired / Commander J. S. EVANS, U. S. Navy
1920 Commander J. S. EVANS, U. S. Navy
1921 Commander J. S. EVANS, U. S. Navy / Commander S. M. ROBINSON, U. S. Navy
1922-23 Commander S. M. ROBINSON, U. S. Navy
1924-25 Commander BRYSON BRUCE, U. S. Navy
1926 Commander A. M. CHARLTON, U. S. Navy
1927 Commander H. B. HIRD, U. S. Navy
1928 Commander H. B. HIRD, U. S. Navy / Captain O. L. COX, U. S. Navy
1929-30 Commander H. T. SMITH, U. S. Navy
1931 Captain O. L. COX, U. S. Navy
1932 Commander H. F. D. DAVIS, U. S. Navy
1933-34 Commander H. B. HIRD, U. S. Navy
1935 Commander C. S. GILLETTE, U. S. Navy
1936 Commander C. S. GILLETTE, U. S. Navy / Commander ROGER W. PAINE, U. S. Navy
1937 Commander ROGER W. PAINE, U. S. Navy
1938 Commander ROGER W. PAINE, U. S. Navy / Lt. Comdr. GUY CHADWICK, U. S. Navy
1939-40 Lt. Comdr. GUY CHADWICK, U. S. Navy
1940-44 Captain J. E. HAMILTON, U. S. Navy
1945 Commander R. T. SUTHERLAND, JR., U. S. Navy
1945-48 Captain F. W. WALTON, U. S. Navy